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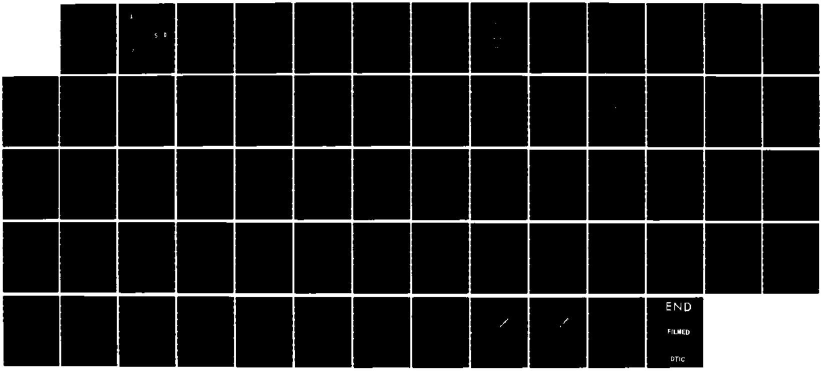

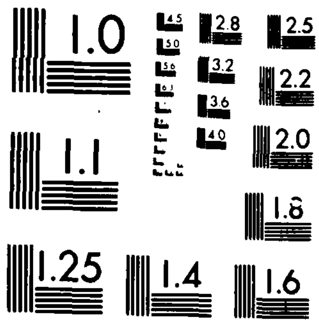

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TECHNICAL REPORT RG-85-22

THE ANALYSIS, DESIGN AND PERFORMANCE OF A 1.25 INCH DIAMETER PNEUMATIC ACTUATOR

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U.S. ARMY MISSILE COMMAND
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This report presents the detailed design and measured performance of a prototype, single axis vane actuator for potential application to small diameter hypervelocity missiles. The actuator produces over 100 inch-pounds of torque while operating on nitrogen at 1295 psi. The small signal bandwidth is approximately 40 Hz. The actuator operates 2 vanes on a common shaft for aerodynamic control in the pitch or yaw plane.

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I. INTRODUCTION

For several years, the Army Missile Command has been interested in exploiting the potential of Kinetic Energy (KE) warheads: solid rod penetrators boosted to high speeds by Hypervelocity Missiles (HVM). In 1968, a special task group invesitgated the problem of accelerating a 10 lb payload to 6500 ft/sec in 0.25 seconds. That program was followed by the development of a Single Penetrator Kinetic Energy (SPIKE) missile. Another program called Solid Propellant Advanced Ramjet Kinetic Energy (SPARK) looked at the problems inherent in guiding a KE missile. Concurrent with these efforts, a Hypervelocity Concept Study Team investigated the advantages of KE warheads against heavily armored targets.

In 1982, a Hypervelocity Test Bed Missile Baseline Design Study looked into five different concepts for guided KE missiles:

1. Discardable Booster

2. Flyaway Booster

3. Boost - Sustain

4. Boost - Coast

5. Solid Boost - Ramjet Sustain

The discardable booster utilized four external motors to accelerate a small diameter centerbody to the required velocity. The spent motors were then separated and the centerbody was guided to the target. The flyaway concept used one large booster motor which separated at burnout allowing the centerbody to coast while guided to the target. The other three concepts were of conventional design where the boost motors were not separated from the warhead.

In the discardable booster and flyaway concepts, control surfaces are required on the centerbody. The control surfaces operate in the plume of the motor(s) during boost and operate as controlled aerodynamic vanes during coast. This report is concerned with the design and evaluation of a pneumatic actuator (see Figure 1), for the control of a 1.65 inch diameter centerbody used in the discardable booster and flyaway HVM concepts. The prototype actuator is 1.25 inches in diameter. For the lowest possible drag, the diameter of the centerbody was the driving constraint for the design of the actuator and other missile components.

The output power levels, small size and short flight time did not favor an electromechanical actuator design. The prototype actuator is designed to operate on nitrogen. If higher response is required, helium could be used. The penalty for using helium is slightly reduced reliability due to smaller orifice sizes and more potential leakage problems during long term storage.

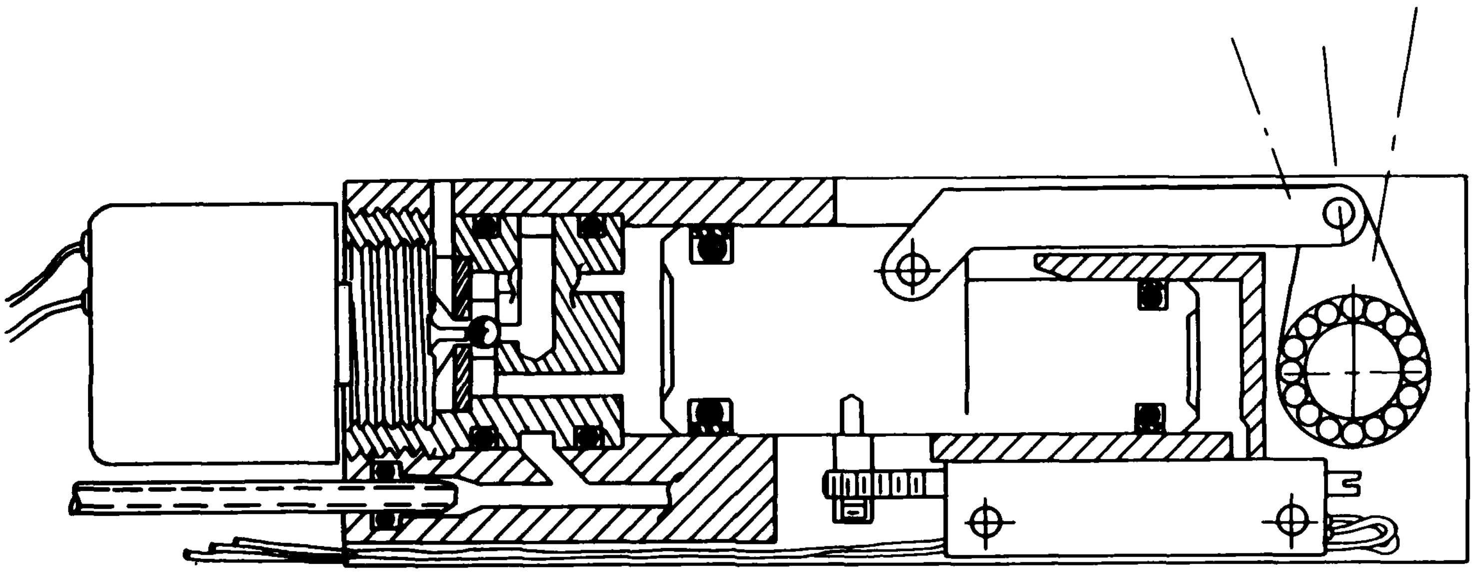


Figure 1. <u>1.25 in dia tangent piston actuator</u>.

II. DESIGN REQUIREMENTS

The actuator was designed to address technical issues concerned with the 2 axis control of a spinning centerbody boosted to about 5000 ft/sec. The 1.25 inch diameter actuator is the third iteration of a design cycle that considered 2.06 and 1.60 inch diameter actuators. The prototype design is feasible only beacuse of the rather generous length allowed for the control section by the large length/diameter ratio of the centerbody vehicle. The small diameter of the centerbody limits the actuator output shaft lever arm length to about 0.5 inches and also limits the size of the solenoids and pistons. Small pistons and high pressures must be used.

The 1.25 inch diameter actuator will fit inside a 1.65 inch outside diameter structural tube with 0.20 inch wall thickness, which is the primary load carrying structure of the centerbody vehcile. One actuator can provide 2 axis control for a spinning vehicle as in STINGER. For a dual axis control section (see Figure 2), 2 single axis actuators can be positioned end-to-end, which will displace the vane axes by 0.62 inches and should not be a problem with such a long vehicle.

The design requirements are listed in Table 1. The critical design driver was the 1.25 inch maximum diameter required for the smallest possible drag on the centerbody. The stall torque required is a fairly arbitrary number since the missile flies at nearly a constant speed which should permit an accurate hinge line location with respect to the vane center of pressure. At the start of the design the bandwidth requirement was not known, but was estimated at 25 Hz minimum. A recent study on an HVM interceptor indicates that a 35 Hz bandwidth is acceptable. The bandwidth requirement is also influenced by the guidance scheme utilized, missile spin rate, update rate, etc.

For a tactical weapon there would be design requirements not listed in Table 1. A weather seal would be required on each vane shaft to protect the actuator bearings from the hot plume gasses during boost; and the misalignment between the vanes on the same output shaft should be held to within 1 mil. The weather seal would be contained within the tube wall thickness.

Actuator assembly in the structure tube is a problem area. The present design could be assembled anywhere within the tube but the prototype vane attachment concept (pinned half lap joint) should be modified for a production version. H. R. Textron has designed and fabricated, under contract to MICOM, a 1.25 inch diameter hydraulic actuator. In their proposal, Textron proposed a splined van attachment concept that would be suitable for a tactical missile.

Because of the anticipated stiffness requirements for the centerbody structural tube, only one joint in the tube was allowed for assembly purposes. The prototype actuator design uses two access holes, one on top and one on the bottom, to gain access to the pinned crank and output shaft joint.

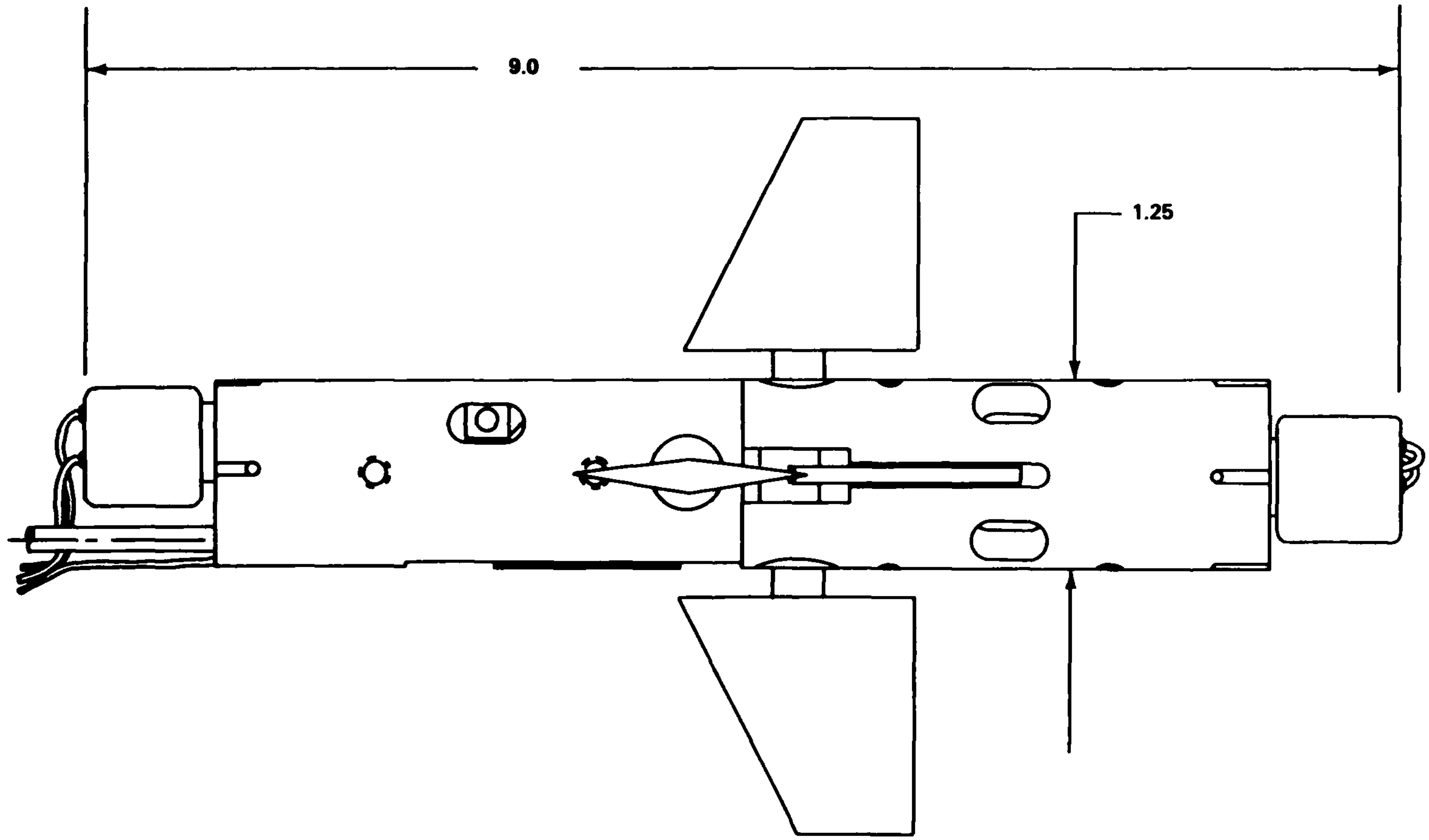


Figure 2. 2 axis control section.

TABLE 1. Design Requirements.

1. Bandwidth; 25 Hz or greater when loaded at 6.67 in-lb/deg per axis with input signal of $\pm$ 2.5 degrees.

2. No load slew rate; 400 deg/sec minimum.

3. Stall torque; 150 in-lb/axis desired, 100 in-lb/axis required.

4. Deflection; $\pm$ 15 deg.

5. Vane inertia; 1.24×10^{-4} lb-in-sec^2 each or 2.48×10^{-4} lb-in-sec^2 per axis.

6. Nominal load; 6.67 in-lb/deg per axis, opposing.

7. Vane panel loads; 93.3 lb lift, 50 lb drag, (see Figure 11).

8. Time of operation; 5 seconds.

9. Diameter; 1.25 inches maximum.

10. Length; 7.0 inches maximum.

11. Weight; 2.0 pounds.

III. DESIGN DESCRIPTION

The prototype actuator is a conventional push-pull dual area piston design using a solenoid operated ball valve and unique tangent piston. The tangent piston and split cylinder arrangement allows the connecting rod to be on the outside of the cylinder, which results in a short and small diamter design. The design also reduces the angular motion on the connecting rod thereby reducing the sine-cosine effects due to piston motion. A gas transfer tube conducts gas into the actuator and also between actuators when used as a 2 axis control section. The ends of the tubes are sealed by "O" rings in groves in the actuator housing. The design must accommodate the end thrust from the pressurized transfer tubes.

The solenoid valve controls the pressure on the large end (control end) of the dual area piston. The smaller or reference end of the piston is always pressurized with the full regulated supply pressure (1295 psig). Since the large piston is approximately twice the area of the small piston, the piston will be in equilibrium at zero load when the pressure on the large end is about one half the supply pressure. Under loaded conditions, the pressure on the large end must be controlled to the proper value to maintain a force balance on the piston. Therefore, the piston will extend or retract according to the pressure on the large end of the piston.

The solenoid valve controls the piston pressure by controlling the position of a 0.0937 inch diameter ball between two fixed orifices, an inlet (charge) orifice and an outlet (vent) orifice. The solenoid is operated in a pulse width modulation (PDM) mode at a carrier frequency of 200 Hz. To maintain the piston at a fixed position, the control pressure must be maintained at a constant pressure (for constant load or torque) by admitting gas or venting gas from the large end of the piston. Therefore, this type of valve design, called open center, uses about the same amount of gas whether the piston is moving or at rest. For short flight times and high vane duty cycles, the open center design is not a disadvantage.

The actuator performance is "tuned" by varying the sizes of the orifices and/or changing the stroke of the ball and solenoid. Performance can also be increased by using compensation in the PDM driving circuits. No compensation was used in the measured data presented in this report.

A standard Bourns linear potentiometer, modified by shortening the shaft and filing the sides, is used to measure the piston position. Due to the small angles involved with the connecting rod and output shaft, the piston position is nearly proportional to vane deflection angle. The feedback scale factor is high due to the short piston stroke and short output shaft lever arm. The mechanical scale factor is 8.63×10^{-3} inches per degree and has not been a problem during performance evaluation.

The actuator uses a Lisk Solenoid from an early version of the TOW actuator. Standard needle bearings are used on the output shaft. The clearance between the shaft and the needle rollers was greater than desired and precision bearings should be used on a flight actuator. Shamban GLYD ring seals are used on the piston. Standard BUNA N "O" rings are used for the static seals on the valve assembly and the gas transfer tube. The piston and cylinder bores were hard anodized. The prototype actuator uses common materials and no parts were heat treated. Any potential seal or "O" ring leakage has not been a problem. The basic characteristics of the prototype actuator are listed in Table 2.

IV. DETAILED DESIGN

A. Actuator Sizing

Before the actuator can be designed, several assumptions must be made. With a maximum diameter of 1.25 inches, and assuming that the output shaft crank will be in the center of the housing and nearly perpendicular to the longitudinal centerline, the maximum lever arm length will be 0.625 inches. The maximum practical length will be 0.5 to allow for a pinned joint between the connecting rod and crank. The stresses on the pinned joint will be calculated later. It is assumed that the output shaft should deflect through $\pm$ 15 degrees.

Many combinations of piston sizes and supply pressures could be used but it is desirable to keep the supply pressure as low as possible. Also, each side of the dual area piston should be a common size so as to use readily available seals, which are usually stocked in sizes of 1/16 inch increments in the sizes required for this application.

Referring to the simplified schematic diagram in Figure 3, which ignores piston side loads from the inclination of the connecting rod (which is about 6 degrees) and also ignores turning moments on the piston due to the pressure forces on each end not being concentric, a curve of approximate piston diameters versus supply pressure can be calculated for preliminary design and layout purposes (see Figure 4). It is also assumed that the dual area piston has a 2:1 area ratio even though it is highly unlikely that could be the case if standard size seals are used. Most of the actuator friction is caused by piston seal friction and side loads and is taken into account by the efficiency factor η. A value of $\eta = 0.92$ allows for 8% friction (a common conservative value for pneumatic actuators).

TABLE 2. Design Performance.

1. Bandwidth; loaded 7.09 in-lb/deg/axis

   63 Hz at ± 1 deg at null

   50 Hz at ± 2.5 deg at null

   26 Hz at ± 5 deg at null

   13 Hz at ± 10 deg at null

   36 Hz at ± 2.5 deg at + 7.5 deg

   50 Hz at ± 2.5 deg at -7.5 deg

   Bandwidth; No load

   72 Hz at ± 2.5 deg at null

2. No load slew rates.

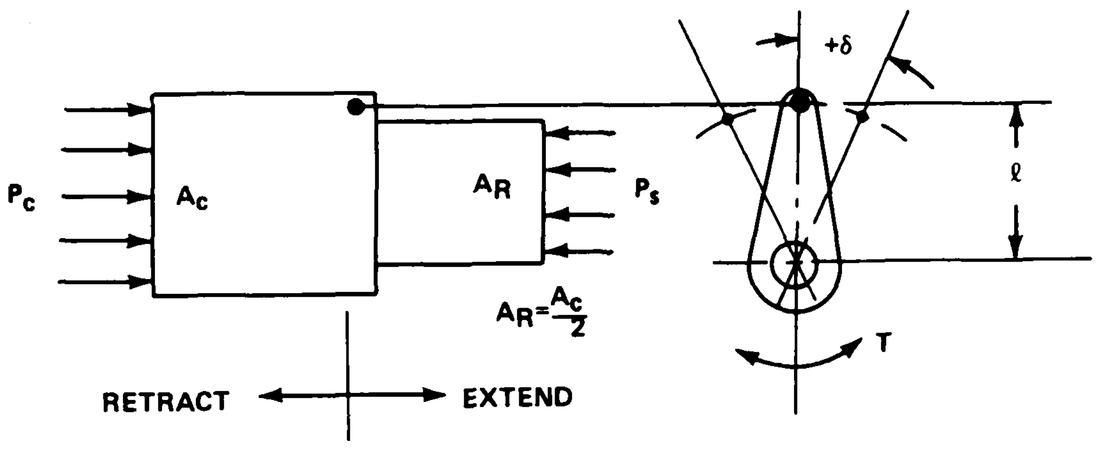
   1092 deg/sec extending

   1000 deg/sec retracting

3. Stall Torque; 128 in-lb

4. Actuator length (without vanes) 4.5 in

5. Actuator Weight (without vanes) 0.56 lb



LEGEND:

T = Output Torque, 100 in lbs Minimum

δ = Vane deflection, $\pm$ 15 deg

A_c = Control piston area, in^2

P_c = Control Pressure, psia

P_s = Supply Pressure, psia

η = Friction factor, 0.92

ℓ = Crank radius 0.5 in

A_R = Reference Piston Area, in^2

Figure 3. Simplified actuator schematic.

Now calculate control piston diameter for a range of supply pressures for T = 100 in lb:

$$\left(P_c A_c - P_s \frac{A_c}{2}\right) \eta \cos \delta = \frac{T}{\ell}$$

All full stall torque in the extend direction, $P_c = P_s$:

$$P_s \left(\frac{A_c}{2}\right) \eta \, \ell \cos \delta = T$$

For layout purposes, piston diameters are more useful than areas for selecting a piston size to fit the required space:

$$P_s \left(\frac{\pi D_c^2}{8} \right) \eta \, \ell \cos \delta = T$$

or: $$D_c = \left(\frac{8T}{P_s \pi \eta \ell \cos \delta} \right)^{1/2}$$ Control Piston Diameter

This equation is plotted in Figure 4 for T = 100 and 150 in-lbs.

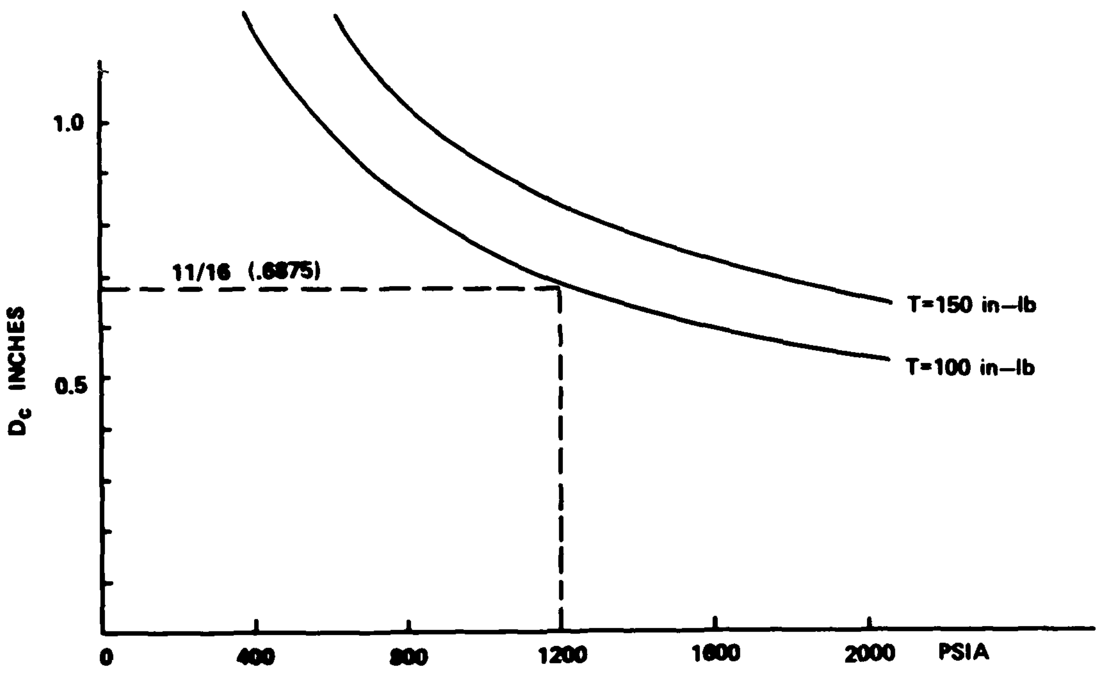


Figure 4. Approximate control piston diameter vs P_s.

Figure 4 is useful for preliminary design purposes.

It is logical to place the solenoid and valve assembly near the control (large) end of the piston. The maximum diameter of the Lisk solenoid is 0.885 inches. The solenoid is placed as close as possible to the "top" edge of the actuator to allow room for the gas transfer tube along the "bottom" edge which also provides the maximum available volume for the feedback pot. If 0.1 inches is allowed for the large cylinder wall thickness, the maximum diameter of the largest possible piston is:

$$D_c = 2\left(\frac{0.885}{2} - 0.1\right) = 0.685 \text{ in}$$

The nearest standard size to this diameter is 11/16 or 0.6875 inches, or an area of 0.371 in^2. The nearest standard diameter for approximately one half that area is 0.500 inches. Therefore, the nominal size of the piston will be 11/16 x 1/2 inches, and the supply pressure will be about 1200 psia as shown in Figure 4. These are reasonable numbers for this design, so the detailed layout can proceed.

The 0.5 inch diameter end of the tangent piston allows space for the connecting link, feedback pot, gas passage hole, and an adequate cylinder wall thickness. It is not absolutely necessary that the solenoid axis be concentric with the valve assembly and large diameter of the piston. However, having these three items aligned produces a compact design and reduces machining time. For assembly purposes, it is necessary that the valve assembly be a larger diameter than the large piston. The piston is installed first, then the valve assembly, then the solenoid.

Once the actuator has been layed out with reasonable dimensions for the various parts, a better estimate can be made for the supply pressure. The nominal sizes of the cylinders, after hard anodizing, are used for the effective piston areas since the pressure acts over the complete seal, not just the piston diameter. The geometry of the arrangement minimizes the sine-cosine effects on the rod and crank. Since the area ration of the piston is not exactly 2:1, the stall torque will be different at the fully extended and retracted position. The worst case (requires highest supply pressure) is extending from the fully retracted position. In this case, control pressure will be equal to the supply pressure:

$$P_s = \frac{T \cos\theta}{(A_c - A_R)\eta\ell\cos\delta} = \frac{100\cos(6.4)}{(0.373-0.198)(.92)(.5)\cos(15)}$$

$P_s = 1278$ psia Where θ is the angle of the connecting rod referenced to the centerline of the piston.

When the piston is fully retracted and against the stops, the full area of the piston is not accessible to the full supply pressure even though the piston head is recessed to prevent such occurrence. It is also desirable to have excess torque, above the stall torque, to accelerate the load inertia when the piston is near the stops. The tests were made at 1295 psig to have some excess torque and produce a conservative design.

B. Orifice Sizing

The ball valve diagram is shown in Figure 5. The unloaded slew rate requirement of 400 deg/sec is used to estimate the size of the upstream charging orifice using nitrogen gas:

$$\dot{\delta} = 400 \text{ deg/s} = 6.98 \text{ Rad/s}$$

The piston velocity $\dot{x}$ is therefore:

$$\dot{x} = \ell\,\dot{\delta} = (0.5)(6.98) = 3.49 \text{ in/s}$$

This velocity is less than the maximum velocity specification for the feedback pot (10 in/sec) and is acceptable. The piston stroke for 15 degrees is:

$$x = \ell\,\delta = (0.5)(15/57.3) = 0.131 \text{ in}$$

NOTE: This is one half the total stroke.

The time required for the piston to move from the fully retracted position to the null position is:

$$\Delta t = \frac{x}{\dot{\delta}} = \frac{0.131}{3.49} = 0.0375 \text{ sec}$$

When fully retracted, the control pressure is 14.7 psia, and the upstream orifice is sealed. When the upstream orifice is opened, choked flow occurs across the orifice while the piston extends. Assume the initial mass of gas in the large cylinder, full retracted, is zero pounds. Actually, there will be a small amount due to the volume of the recessed piston face. After the piston has moved to mid position, assume the pressure has risen to the full supply pressure. The mass of gas in the large cylinder is:

$$w = \frac{P_s V}{RT} = \frac{(1278)(144)\left[\dfrac{(0.373)(0.131)}{1728}\right]}{55.2(530)} = 1.78 \times 10^{-4} \text{ lb}$$

The maximum flow rate is:

$$\dot{w} = \frac{w}{\Delta t} = \frac{1.78 \times 10^{-4}}{0.0375} = 4.75 \times 10^{-3} \text{ lb/sec}$$

The upstream orifice area is:

$$A_u = \frac{\dot{w}\sqrt{T}}{0.53\, P_s C_d} = \frac{4.75 \times 10^{-3}\sqrt{530}}{0.53(1278)(0.65)} = 2.48 \times 10^{-4} \text{ in}^2$$

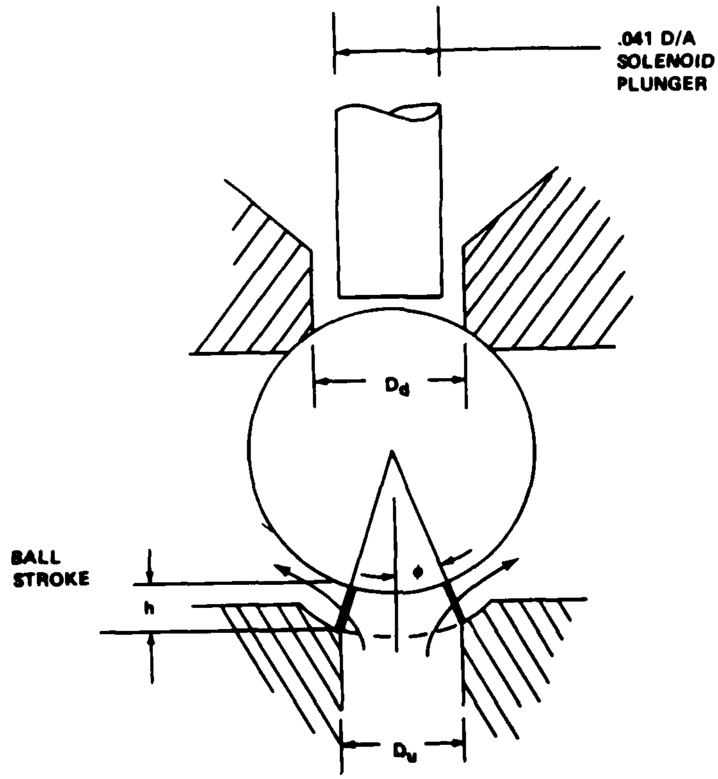


Figure 5. Ball valve diagram.

The upstream orifice curtain flow area is a function of ball diameter, ball and orifice diameter ratio, ball stroke, and discharge coefficient. The initial ball stroke was assumed to be h = 0.006 inches:

$$A_u \approx \pi D_u h C_d \cos 15$$

or

$$D_u \approx \frac{A_u}{\pi h C_d \cos 15}$$

A_u = Upstream orifice area, (in^2)

h = Ball stroke, (.006 in)

C_d = Discharge Coefficient (0.65)

COS 15 allows for "tilting" of the curtain towards the ball

$$D_u \approx \frac{2.48 \times 10^{-4}}{\pi (0.006)(0.65) \cos 15} = 0.021 \text{ in}$$

On the retract stroke, the discharge orifice has choked flow for nearly the whole stroke, and the vent area should be twice the charging area:

$$A_d = 2\ A_u = 2(2.48 \times 10^{-4}) = 4.96 \times 10^{-4}\ in^2$$

The downstream orifice diameter is approximately:

$$D_d \approx \frac{A_d}{\pi\ h\ C_d\ \cos 30} = \frac{4.96 \times 10^{-4}}{\pi\ h\ (0.65)\ \cos 30}$$

$$D_d = 0.047\ in$$

This orifice diameter may be acceptable since the plunger on the TOW solenoid is .041 inches in diameter. Also, the ball diameter should be carefully chosen in relation to the orifice diameters and plunger size. If an orifice seat makes contact on the ball at a "latitude" less than about 45 degrees, there will be sticking problems. In this case, the orifices are so small that the opposite problem occurs; the ball is not captured by the orifice seats.

There is a minimum diameter of downstream orifice where the annular area between the plunger and orifice diameter will be equal to the required area:

$$A_d = \frac{\pi\ D_d^2}{4} - \frac{\pi\ D_p^2}{4}$$

where D_p = plunger diameter = 0.041 in

or

$$D_d = \sqrt{\frac{4}{\pi}\left(A_d + \frac{\pi\ D_p^2}{4}\right)} = \sqrt{\frac{4}{\pi}\left(4.96 \times 10^{-4} + \frac{\pi\ (0.041)^2}{4}\right)}$$

$$= 0.048\ in$$

Therefore, the absolute minimum downstream orifice diameter should be 0.048 inches.

However, this diameter is not much larger than the plunger and provides only .0035 inches of diametrical clearance between the orifice and the plunger. If the downstream orifice diameter is increased to about 0.059 inches, the flow area can be measured graphically as $8.8 \times 10^{-4}\ in^2$ at a stroke of 0.006 inches. The annular area between the plunger and orifice diameter should be checked to make sure the flow area is not restricted:

$$\pi\ \frac{(0.059)^2}{4} - \pi\ \frac{(0.041)^2}{4} = 14.1 \times 10^{-4}\ in^2$$

This annular area is 1.6 times larger than the discharge orifice area and is acceptable, but a larger area would be desirable.

Since the plunger and vent orifice relationship dictated a larger discharge area, the inlet orifice must be increased. The new upstream orifice area will be one half the vent area.

$$A_u = 4.4 \times 10^{-4} \text{ in}^2$$

or:

$$D_u = \frac{A_u}{\pi \, h \, C_d \cos 15} = \frac{4.4 \times 10^{-4}}{\pi \, (.006)(1) \cos 15} = 0.024 \text{ in}$$

NOTE: Since it is easy to enlarge an orifice, and difficult to reduce it, a $C_d = 1$ was used here to give the smallest initial diameters.

Of course, the result of using these larger orifices is that the no load slew rate will be greater than 400 deg/sec. Since the actuator will be tuned by varying the ball stroke and/or changing the orifice sizes, these orifice diameters are reasonable. These sizes could be refined by using more accurate expressions for the flow geometry involved. However, by using assumed values for the discharge coefficients, such second order effects would be masked anyway. Some designers double the flow area as calculated here using the no load slewing rate; but with a ball valve, the flow area is easily adjusted by changing the ball stroke. Therefore, the actuator was built with

$$D_u = 0.026 \text{ inches}$$

$$D_d = 0.059 \text{ inches} \quad .$$

A large scale 100:1 drawing of a 3/32 ball and these seats indicated that the ball would seat on the orifices at about the proper "latitude" to minimize sticking problems. The layout also indicated that a ball guide or cage would not be required because of the short ball stroke and the size relationship between the ball and seats.

Past experience has shown that discharge coefficients can vary from a low value of about 0.61 to 1.0 for small orifices as used here. For ball strokes less than 0.006 inches, the discharge coefficient is unknown.

C. Solenoid Valve

The TOW solenoid valve is not the optimum design for this prototype actuator. Guidance and Control Directorate had several earlier versions of TOW actuators on hand, and since only two or three solenoids would be required, one TOW actuator supplied four solenoids.

In the ball valve assembly, the solenoid return spring forces the ball into the upstream seat, opposing the opening force of the supply pressure integrated over the exposed area of the ball. When the solenoid is energized, the plunger must pull in by overcoming the preload on the return spring while being aided by the pressure forces on the ball. In the TOW application, the opening force on the ball is about 7.2 lbs. That is much more force than required for the prototype actuator.

For a worst case condition, assume the upstream orifice seat is 0.010 inches wide and the ball contacts the outside edge of the seat. Therefore, the effective seat diameter will be 0.026 + .010 + .010 = .046 inches. The opening force of the supply pressure on that area will be:

$$F_{Ps} = P_s\ A = (1309.7)\ \pi \frac{(.046)^2}{4} = 2.18 \text{ lb}$$

On ball valves of this type, the net contact force is on the order of 1 to 2 lbs. The minimum required ball contact force can be estimated from the supply pressure and assumed worst case (largest) seat. This assumes that the ball is in perfect contact with a spherical seat 0.010 wide as shown in Figure 6. We calculate the net force on the ball to give a contact pressure equal to or greater than P_s:

$$F_{net} = P_s\ A_s = P_s \left[\frac{\pi\ (0.46)^2}{4} - \frac{\pi\ (.026)^2}{4} \right]$$

$$F_{net} = F_{spring} - F_{Ps}$$

or $$F_{spring} = F_{net} + F_{Ps}$$

$$= 1.48 + 2.18$$

$$F_{spring} = 3.66 \text{ lb MINIMUM}$$

As assembled in the prototype actuator, the plunger preload was about 7.98 lbs, which should allow the valve to withstand 300 g's. The maximum plunger mass is 0.0075 lbs. which becomes 2.25 lbs at 300 g's.

At the end of the charge cycle (piston extending), the solenoid return spring must overcome the seating force of the supply pressure, reduced due to the pressure drop across the charge orifice, acting over the exposed area of the ball. As in the previous case, assume a seat width of 0.010 inches D_d = 0.059 inches, fully supply pressure of 1309.7 psia and that the ball contacts the seat on the outside edge. Therefore, the effective seat diameter is 0.079 inches.

$$F = P_s\ A = (1309.7)\ \pi \frac{(.079)^2}{4} = 6.34 \text{ lb}$$

The net force of the return spring is greater than this value so a ball lockup condition will not occur.

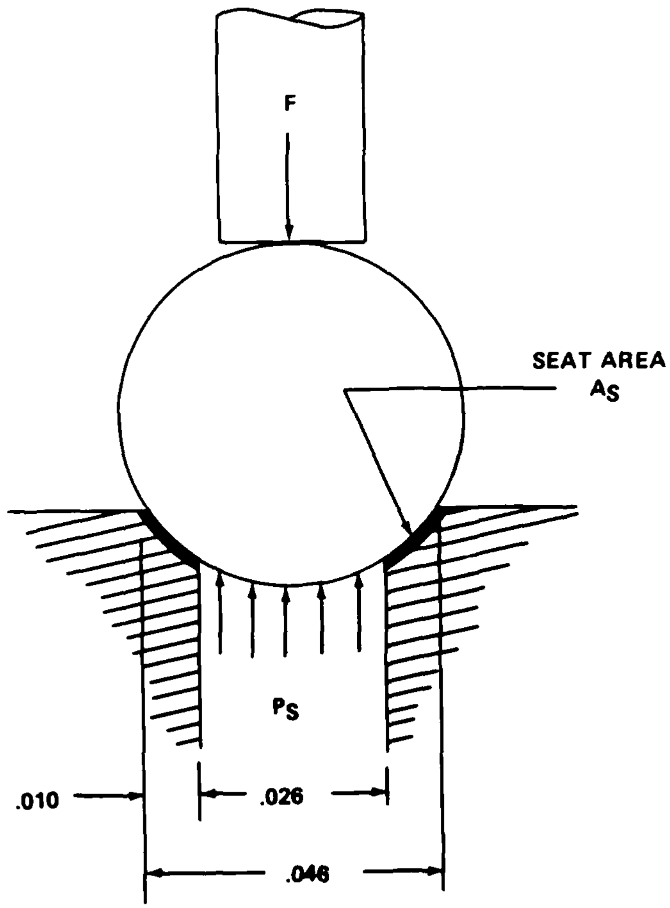


Figure 6. Seat contact area.

The valve lockup pressure may be estimated by using the assembled (compressed) return spring force and the same effective downstream area:

$$\text{Lockup Pressure} = \frac{7.98}{\frac{\pi\,(.079)^2}{4}} = 1628 \text{ psi}$$

When the solenoid spring is adjusted for the best actuator performance, the actuator will operate between supply pressures of 975 to 1600 psig. Since the actuator was only proof tested to 2000 psig, the lockup pressure test was terminated at P_s = 1600 psig.

From this discussion it can be appreciated that the optimum solenoid design is dependent on many juggled factors associated with design of two orifice ball valve; mainly orifice diameters, seat contact widths, supply pressure, solenoid pull in force, plunger spring constant and preload, etc. For this actuator, the nominal seat width was 0.005 inches for both orifices.

D. Solenoid Vent Pressure

The TOW solenoid valve is sealed with epoxy and rated at the following pressures:

Working = 50 psig

Proof = 75 psig

Burst = 125 psig

Precautions must be taken to insure that the solenoid is not exposed to more than 50 psig. A detailed analysis of the pressure drops downstream from the discharge orifice was made. The restrictive areas are shown in Figure 7. All of the areas are expressed to 10^{-3} power so they can be compared directly. It can be seen that area 5, the thread relief groove, is restrictive when compared to the other flow paths. This will cause excessive pressure to build up on the solenoid. The discharge orifice seat area of 0.88 x 10^{-3} in^2 corresponds to a 3/32 diameter ball, stroke of 0.006 inches, and orifice diameter of 0.059 inches.

Assume the stroke could increase to 0.008; therefore, the maximum seat area will be 1.18 x 10^{-3} in^2. The maximum discharge flow rate at 1295 psig (1309.7 psia) will be:

$$\dot{W}_d = \frac{0.528\ P_s\ A_d}{\sqrt{T}} = \frac{0.528(1309.7)(1.18 \times 10^{-3})}{\sqrt{530}} = 0.035 \text{ lb/s}$$

This is extremely conservative because the downstream orifice can never be exposed to the full supply pressure due to the pressure drop across the upstream orifice. However, adequate vents can be provided for even this impossible flow rate.

Calculate a vent area required to flow 0.035 lbs/sec at a maximum source pressure of 50 psig (64.7 psia); the maximum working pressure of the solenoid.

Vent Area: $$A_v = \frac{\dot{W}_d \sqrt{T}}{0.528\ P} = \frac{(.035)\ \sqrt{530}}{(.528)(64.7)}$$

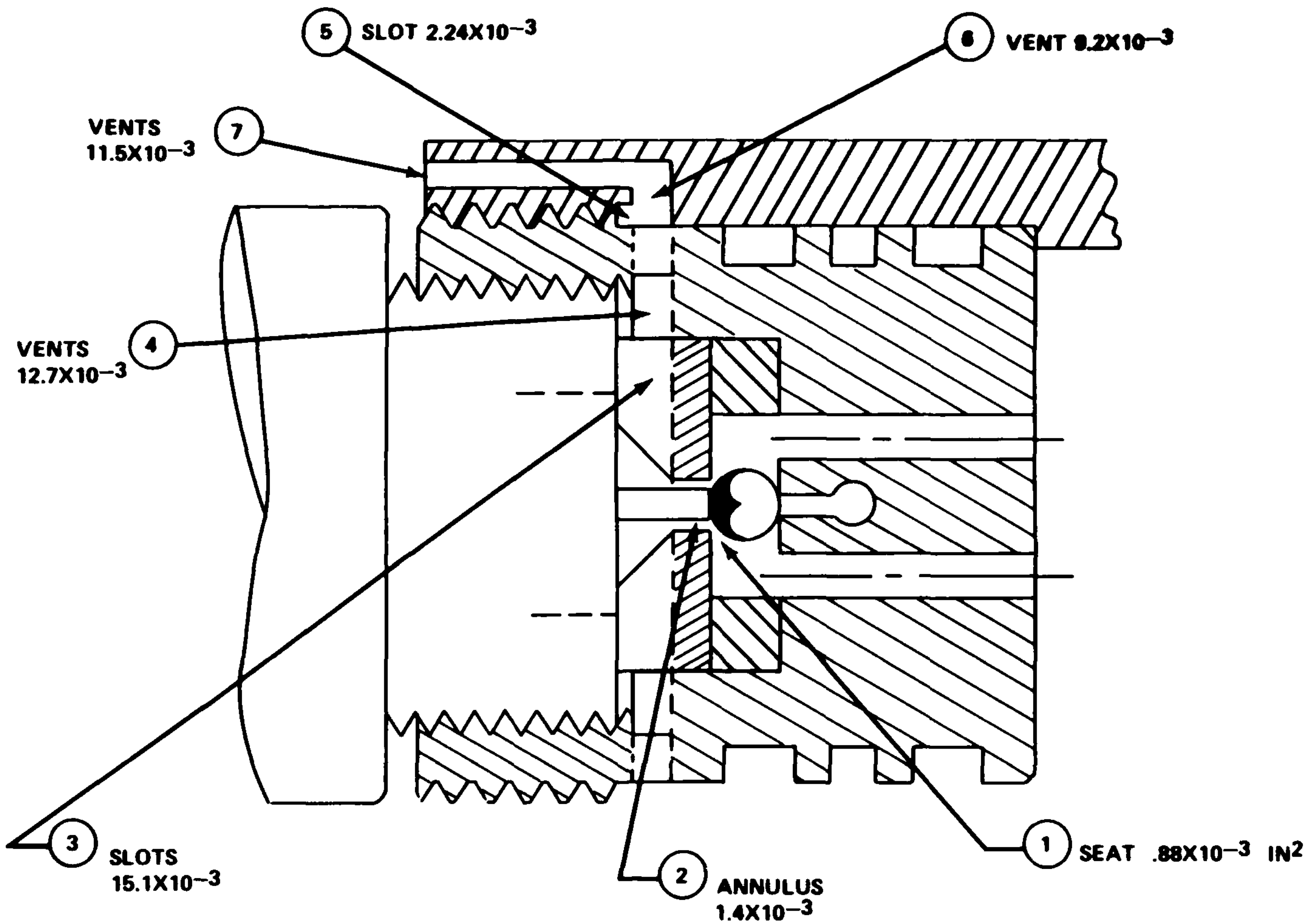


Figure 7. Restrictive areas downstream of Ad .

This vent area can be provided by milling four slots in the solenoid screw thread. Each slot is 0.54 inches deep by 0.110 inches wide. The strength of the solenoid threads in tension is adequate with these four slots.

$$A_v = 4(.054)(0.11) = 23.8 \times 10^{-3}\ in^2$$

The three smaller vent slots in the actuator housing provide additional venting area.

The assembled actuator was proof tested to 2000 psig for 1 minute in both the fully extended and retracted positions. No problems were observed.

E. Piston Design

A diagram of the tangent piston is shown in Figure 8. The rod can be in tension or compression. The connecting rod angle θ varies from 5.3 degree retracted to 6.4 degree extended. Use θ = 6.4 degree for all cases (worst case):

$$F_x = F \cos \theta$$

$$F_y = F \sin \theta$$

Since the line of action of the rod is perpendicular to the crank, and since the rod inclination varies by a small angle, the effective crank lever arm is the same length at both the extended and retracted position:

$$\ell' = \ell \cos \delta = (.5) \cos 15 = 0.493 \text{ in}$$

Therefore, the force in the rod will be:

$$F = \frac{T}{\ell'} = \frac{100 \text{ in-lb}}{0.493 \text{ in}} = 202.8 \text{ lb (NO FRICTION)}$$

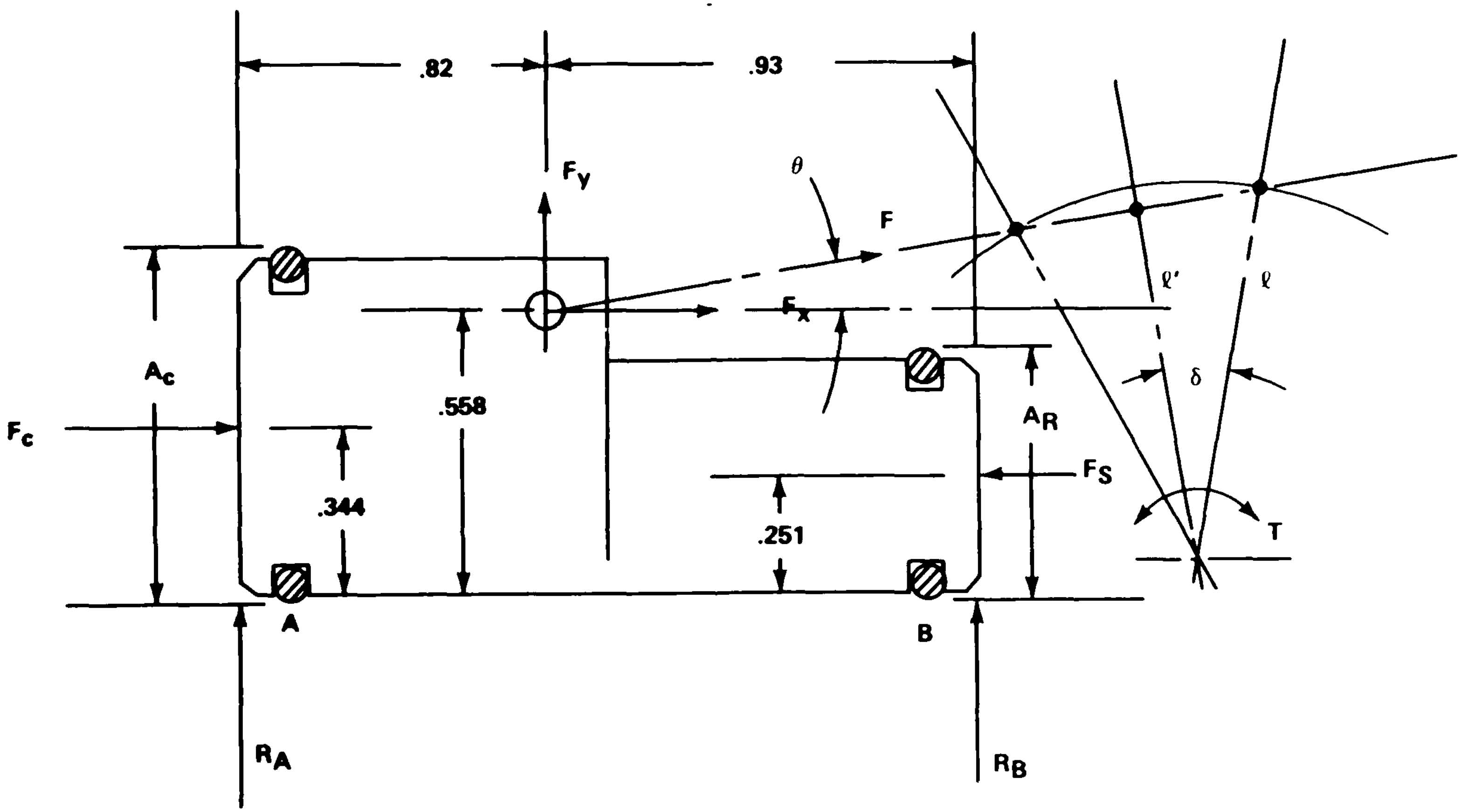
A 2% friction and 20% fitting factor should be used for analysis. Therefore, the connecting rod load will be:

$$202.8\ (1.22) = 247 \text{ lbs force}$$

$$F_x = 247 \cos 6.4 = 245 \text{ lbs}$$

$$F_y = 247 \sin 6.4 = 27.5 \text{ lbs}$$

For the analysis, the geometry of the problem is considered fixed, but the pressure forces do change, and cause moments on the piston because the piston axes are not concentric:



A_C = .373 IN^2 $F_C = P_C A_C$ = 409 LB OR 5.5 LB
A_R = .198 IN^2 $F_S = P_S A_R$ = 259 LB
F = ROD FORCE = ± 247 LB

Figure 8. Tangent piston diagram.

CASE I $P_c = P_s = 1309.7$ psia with rod in compression at T = 100 in-lb (EXTENDING)

$$\overset{+}{\curvearrowright} \sum M_A = 0$$

$$489(.344) + 27.5(.82) - 245(.558) - 259(.251) - R_B (1.75) = 0$$

$$R_B = -6.2 \text{ lb}$$

$$\overset{+}{\curvearrowright} \sum M_B = 0$$

$$R_A(1.75) + 489(.344) - 27.5(.93) - 245(.558) - 259(.251) = 0$$

$$R_A = 33.8 \text{ lb}$$

$$\overset{+}{\uparrow} \sum F_y = 0$$

$$33.8 - 6.2 - 27.5 = 0$$

CASE II $P_c = 14.7$ psia with rod in tension at T = 100 in-lb (RETRACTING)

$$\overset{+}{\curvearrowright} \sum M_A = 0$$

$$5.5(.344) - 27.5(.82) + 245(.558) - 259(.251) - R_B(1.75) = 0$$

$$R_B = 29.2 \text{ lb}$$

$$\overset{+}{\curvearrowright} \sum M_B = 0$$

$$5.5(.344) - R_A(1.75) + 27.5(.93) + 245(.558) - 259(.251) = 0$$

$$R_A = 54.9 \text{ lb}$$

$$\overset{+}{\uparrow} \sum F_y = 0$$

$$-54.9 + 27.5 + 29.2 = 1.8 \approx 0 \text{ lb}$$

The piston forces while extending and retracting are summarized in Figures 9 and 10.

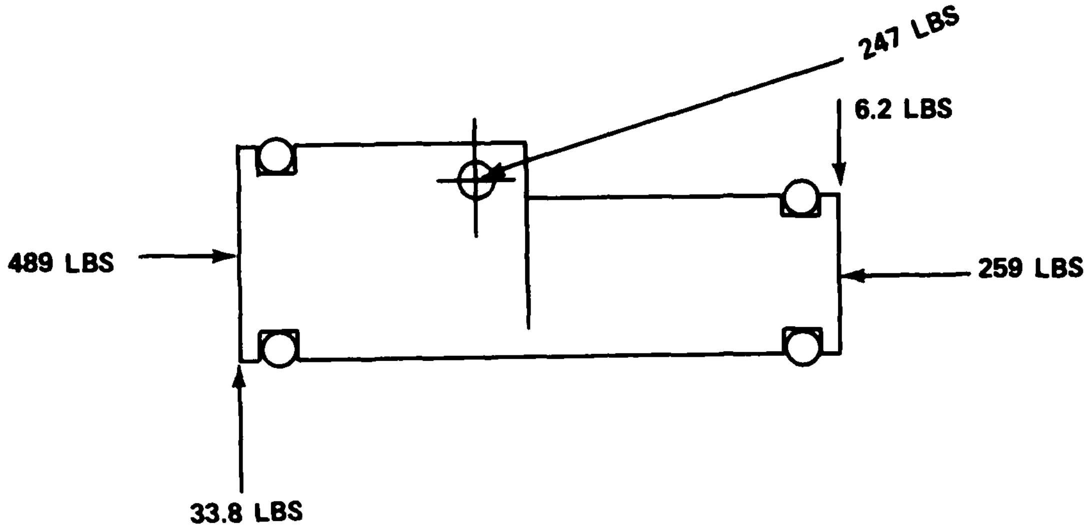


Figure 9. Piston freebody diagram, extending.

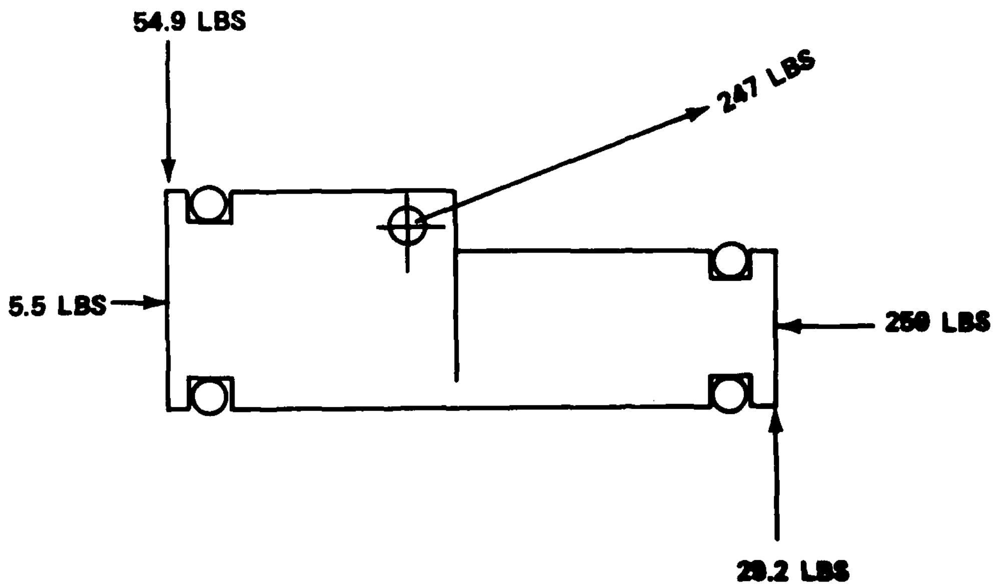


Figure 10. Piston freebody diagram, retracting.

While these side forces appear large, they are in the range of side forces commonly used in conventional piston/rod/crank arrangements. For example, in one large industrial engine, the maximum piston side load was 15.9% of the maximum connecting rod load.

In the present example, the maximum side load is about 22% of the rod load and has not been a problem. The side load could have been reduced by using a longer piston. The seals used on the piston are quite stiff and the piston does not appear to make metal contact with the cylinder wall. The piston was designed with enough length to incorporate a Shambam "Glyd Ring" bearing if the side loads were too large, which was not the case.

For a flight actuator, the piston should be lightened to reduce the amount of boost propellant required and to reduce inertial loads on the output shaft during boost. The sides of the piston between the seals could be milled off. Another method is to drill a 0.312 diameter hole 0.5 inches deep into the large piston face. This method would also tend to equalize the response between the extended and retracted positions due to the large volume caused by the very short stroke. In the present design, the large cylinder volume at +12.5 degree is 10.9 times the volume at the -12.5 degrees position. With the 0.312 "lightening" hole, the volume ratio would decrease to 2.76 to 1. This would decrease the bandwidth slightly at all positions, but more so at the retracted positions, thus equalizing performance, because in the present configuration, the retracted bandwidth is about 30 Hz greater than in the extended position for small input signals.

The 3/32 diameter piston pin and hole were checked for shear, bearing, tear out and tension failure.

Shear: 3/32 diameter (.094) piston pin in double shear.

$$\sigma = \frac{247}{\frac{2\pi(.094)^2}{4}} = 17{,}796 \text{ psi}$$

an acceptable shear stress for steel

Bearing: Piston pin bearing stress in aluminum piston.

$$\sigma = \frac{247}{2(.11)(.094)} = 11{,}943 \text{ psi}$$

low bearing stress for 6061-T6 AL

Tearout: Piston failure due to shearing of aluminum:

$$\sigma = \frac{247}{4(.133)(.14)} = 3{,}316 \text{ psi}$$

low shearing stress

Tension: Assume 247 lbs force acts on only the piston cross section area above pin hole.

$$\sigma = \frac{247}{(.133)(.08)} = 23{,}214 \text{ psi}$$

a low tensile stress for 6061-T6 AL.

The piston seal flange was analyzed for bending. An element of the flange was considered as a small cantilevered beam with the prorated seal force acting on the outer edge. The piston flange bending stress was very low at

σ = 7,222 psi   a low flange bending stress

The piston and cylinder bores were hard anodized. Assuming that hard anodizing builds up the surface about 0.001 inches, the piston to bore clearance was held to 0.001 to 0.006 inches. Shamban GLYD Ring seals are used on each diameter of the piston.

F. Connecting Rod

The rod or link operates in tension and compression, with additional bending stresses due to the bent end. Since the bend is close to one end, bending stresses were not analyzed. The shear stresses on the pins are the same as calculated for the piston.

Bearing: $$\sigma = \frac{247}{(.094)(.125)} = 21{,}021 \text{ psi}$$

Tearout: $$\sigma = \frac{247}{2(.047)(.125)} = 21{,}021 \text{ psi}$$

Tension: $$\sigma = \frac{247}{2(.047)(.125)} = 21{,}021 \text{ psi}$$ low stresses for 340 SS

G. Lever Arm

The lever arm, or crank, was analyzed for a 247 lb tension or compression load.

Bearing: $$\sigma = \frac{247}{2(.079)(.094)} = 16{,}630 \text{ psi}$$ low stress

Tearout: $$\sigma = \frac{247}{4(.047)(.079)} = 16{,}630 \text{ psi}$$

Tension: $$\sigma = \frac{247}{2(.047)(.079)} = 33{,}216 \text{ psi}$$

These stresses are all acceptable for 304 SS. As would be expected, the highest stress is due to tension on the "eye" of the crank. This is a direct result from trying to achieve the longest possible lever arm by placing the hole as close to the crank end as possible.

H. Output Shaft

The diameter of the output shaft should be as large as possible to keep the vane deflections as small as possible. The estimated vane panel loads are shown in Figure 11. The pure twisting load is one half the stall torque. The lift and drag forces are combined vectorially to get a shaft bending moment of:

$$M = \sqrt{(93.3)^2 + (50)^2}\ (0.8) = 84.7 \text{ in-lb}$$

Assuming an allowable shear stress σ_{SAL} of 25,000 psi the shaft section modulus should be:

$$\frac{\pi d^3}{16} = \frac{\sqrt{T^2 + M^2}}{\sigma_{SAL}} = \frac{\sqrt{(100)^2 + (84.7)^2}}{25{,}000} = 5.24 \times 10^{-3} \text{ in}^3$$

or: $d = 0.299$ inches.

The prototype actuator uses a 0.312 output shaft. The shaft should be checked for angular twist due to a torque equal to one half the stall torque:

$$\phi = \frac{TL}{GJ}$$

$T = 50$ in lbs.

$L = .825$ in (shaft length, $\frac{1.65}{2}$)

$G =$ shear modulus, 11×10^6 psi

$$J = \frac{\pi d^4}{32} = 9.36 \times 10^{-4} \text{ in}^4$$

$$\phi = \frac{50(.825)}{(11 \times 10^6)(9.36 \times 10^{-4})} = 4.00 \times 10^{-3} \text{ Rad} = .23 \text{ deg}$$

This amount of shaft twist may not be acceptable for a flight actuator. If shaft stiffness is critical, an allowable shear stress of about 8,000 psi should be used:

$$\frac{\pi d^3}{16} = 5.24 \times 10^{-3} \left(\frac{25{,}000}{8{,}000}\right) = 1.64 \times 10^{-2} \text{ in}^3$$

or: $d = 0.437$ in (or 7/16 dia)

The output shaft could be increased to 0.4375 by using precision bushings rather than needle bearings. The larger output shaft twist when loaded at 50 in-lb would be:

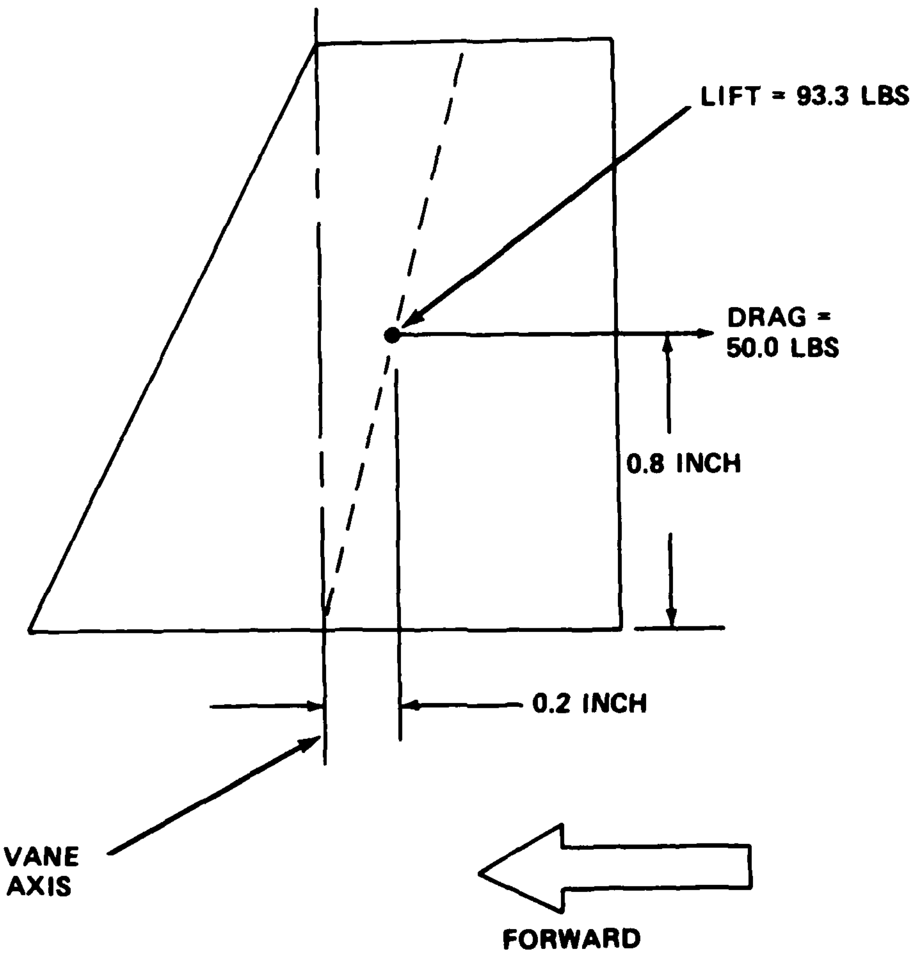


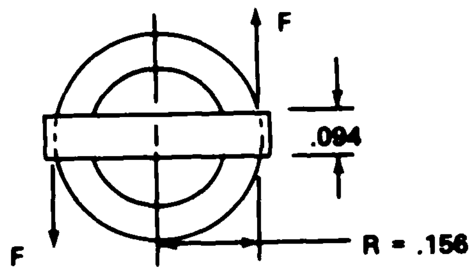
Figure 11. Estimated panel loads.

$$\phi = \frac{TL}{GJ} = \frac{50(.825)}{(11 \times 10^6)(3.597 \times 10^{-3})} = 1.04 \times 10^{-3} \text{ RAD}$$

$$= .059 \text{ DEG} \approx 1 \text{ MIL}$$

The actual vane deflection angle due to shaft twisting could probably be kept to a value slightly lower than this by careful design of the vane, shaft and crank interface.

In the prototype actuator, each fin has an integral shaft with a half-lap joint at the end. A 3/32 diameter dowel pin is used to pin the two fin shafts and the crank together. This method requires a small (0.5 in) hole in the structure tube to provide access to the joint. The pin is in single shear at two shear planes. One half the stall torque is resisted by each shear plane:



$$F = \frac{T/2}{R} = \frac{50 \text{ in-lb}}{.156} = 320.5 \text{ lb}$$

$$\sigma = \frac{F}{A} = \frac{320.5}{\frac{\pi (.094)^2}{4}} = 46{,}183 \text{ psi}$$

This is a fairly high stress but acceptable.

Standard commercial grade Torrington needle bearings were used for the output shaft. These bearings had more radial clearance than anticipated, probably due to less squeeze than recommended at installation, and a flight actuator should use double precision bearings, or a thin wall bushing.

I. Position Feedback Potentiometer

The piston position is measured by a standard Bourns Instruments Model 141 linear motion transducer (see Figure 12). The mechanical stroke is 0.5 inches, the electrical stroke is 0.31 inches and the nominal resistance is 5K ohms. This tranducer fits very nicely into a slot along the bottom of the actuator housing. This commercial grade (not MIL qualified) pot has continuous resolution with 2% linearity. The pot is attached to a small pin that is pressed into a reamed hole in the bottom of the piston. A clearance slot for the pin is milled through the bottom of the cylinder. The body of the pot is riveted and the rivet heads must be filed slightly to fit into the slot. The pot shaft must be shortened. In a flight actuator, the pot position would be reversed and the lead wires would be routed forward through the slot provided on the bottom of the housing. The high position scale factor of 8.73 x 10^{-3} inches per degree of vane deflection has not been a problem.

J. Ball Valve Assembly

The ball valve assembly consists of the solenoid, downstream orifice, spacer, ball, and solenoid adapter, which contains the upstream orifice and also serves as a housing for this assembly. The solenoid screws into the solenoid adapter, clamping the orifices together with the spacer and ball in between. The ball valve assembly screws into the actuator housing and is sealed by two Buna N "O" rings. The valve assembly is accurately located in the actuator housing by a shoulder, which defines the piston stroke in the retracted position and serves as a mechanical stop. A groove around the solenoid adapter between the "O" rings indexes with a hole in the actuator housing which connects to the gas inlet tube. A through hole, drilled across the adapter through the groove, intercepts the reamed upstream orifice hole. Therefore, inlet gas can be admitted to ball cavity between the orifices.

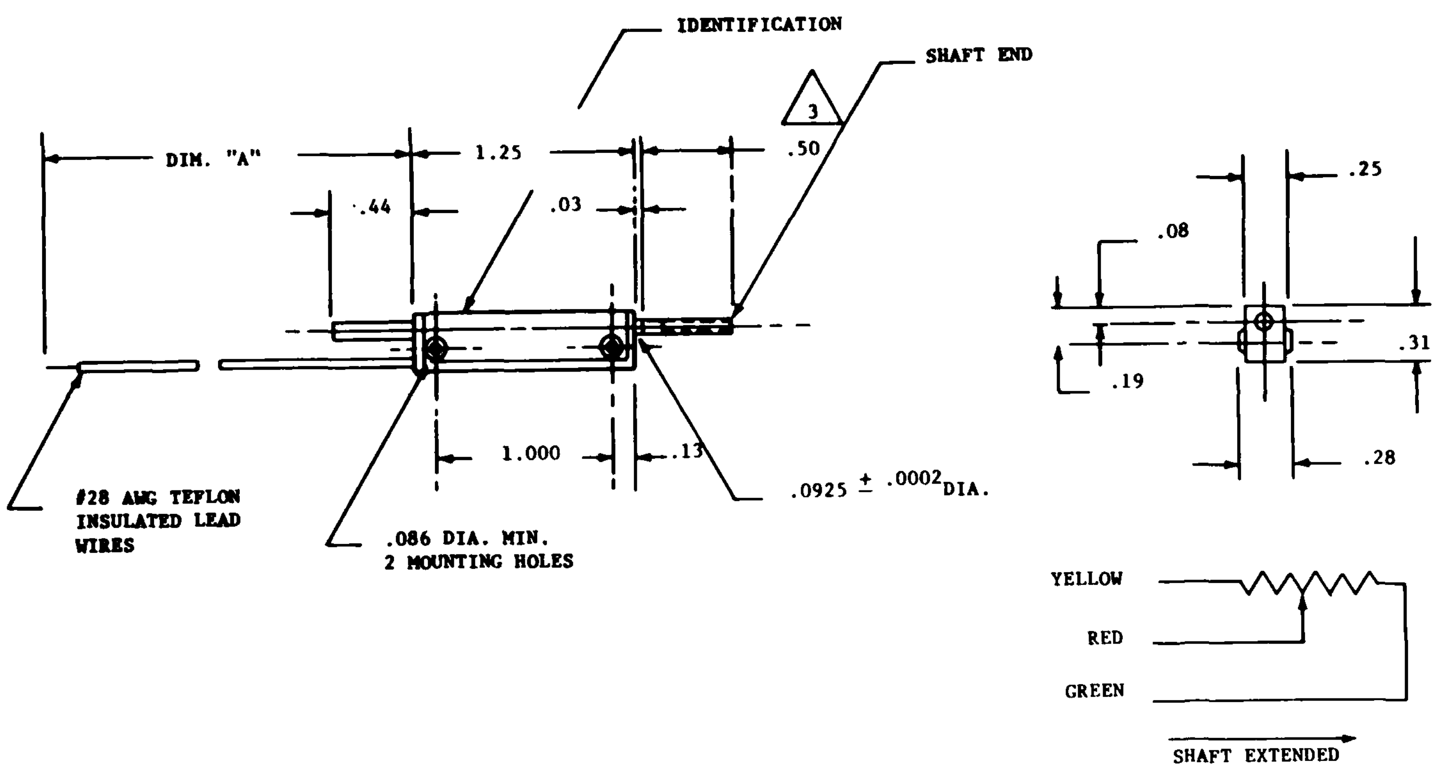


Figure 12. Linear motion potentiometer.

Gas may be vented from the ball cavity through the downstream orifice and out into the missile interior. The large piston is connected to the ball cavity by two holes through the adapter. Therefore, the ball position controls the pressure in the ball cavity and also in the large piston cylinder. The ball stroke is controlled by the thickness of the spacer in the ball cavity. The ball stroke is easily adjusted by adding shims on either side of the spacer.

The initial orifice seat widths were 0.005 inches, formed by cutting with a 3/32 ball end mill and polishing to a 16 finish with a 3/32 diameter ball. The compressive stress on the seats is fairly high, especially on the smaller upstream orifice. The forces are difficult to analyze due to the small dimensions and motions involved. The stiatic compressive stress on the upstream orifice seat is on the order of 16,000 psi and the dynamic stress from the ball impulsive forces is certainly higher:

$$\sigma = \frac{\text{SPRING LOAD}}{\text{SEAT AREA}} = \frac{7.98 \text{ lb}}{4.87 \times 10^{-4} \text{ in}^2} = 16,396 \text{ psi}$$

During the first hours of testing, the upstream orifice seat did "pound out," which increased the ball stroke by about 0.004 inches. The orifice stabilized after that, apparently due to work hardening of the seat. The stress levels on the solenoids and adapter threads due to pressure forces were analyzed and found to be acceptable.

K. Actuator Housing

The housing is machined from 6061-T6 aluminum. The diameters for the solenoid adapter and 11/16 end of the piston are bored 0.125 inches off center. The 1/2 inch diameter for the small end of the piston is bored 0.031 inches off center. The cylinder bores were hard anodized. The housing has an upper slot for the connecting rod, a wide lower slot for the feedback pot and a small bottom groove for wires to pass through. The housing contains a longitudinal, deep hole that indexes with the gas transfer tube at assembly and conducts gas to the ball valve and the bottom end of the small cylinder.

The entrance to the small cylinder is conical shaped to prevent seal damage when the piston is installed. The cone is bored off center as a ramp is not required where the two cylinder bores are tangent. A thin sleeve with a tapered bore is used to replace the valve at assembly to guide the large piston seal across the shoulder where the valve seats in the housing.

The housing is not highly stressed and could be slightly lighter in weight for a flight application. Note that the housing does not have to withstand hoop stresses in the region between the piston seals. Also, due to the short stroke, the length of cylinder bores exposed to hoop stress is very short. The housing was checked for hoop stress at three diameters:

Small cylinder, $\sigma = \frac{P_s r}{t} = \frac{(2014.7)(.253)}{.065} = 7,817 \text{ psi}$

Large cylinder,

$$\sigma = \frac{(2014.7)(.346)}{.151} = 4{,}616 \text{ psi}$$

Solenoid Adapter Bore (.750 diameter),

$$\sigma = \frac{(2014.7)(.376)}{.121} = 6{,}260 \text{ psi}$$

These hoop stresses are low even for 6061-T6 which has a yield strength of about 33,000 psi min.

The gas inlet hole is located in a plane rotated 45 degrees from a "vertical" longitudinal plane defined by the centerlines of the two cylinders. The offset location permits a second actuator to be joined to the first one; their output shafts at 90 degrees to each other. Considering the packaging of only one actuator, the offset inlet is still required so as to pass beside the potentiometer on the way to the end of the small cylinder.

The housing does contain a highly stressed area around the recessed "O" ring groove at the gas inlet. The internal threads for the ball valve assembly come very close to the "O" ring groove. Also, the "O" ring groove comes close to the outside edge of the housing.

Hoop stress, inside edge;

$$\sigma = \frac{P_s r}{t} = \frac{(2014.7)(.125)}{.0127} = 19{,}829 \text{ psi}$$

This stress is unrealistic because the valve assembly will support the aluminum in this area. At the outside edge of the inlet gas "O" ring groove, the hoop stress is:

$$\sigma = \frac{(2014.7)(.125)}{.0355} = 7{,}094 \text{ psi}$$

These stresses are based on worst case wall thickness and proof pressure and should be conservative. The housing was not analyzed for bending and tension loads because, by inspection, the stresses would be very low.

The gas inlet fitting is a "pitot" tube 0.156 inches in diameter with a rounded nose and 0.099 inch diameter through hole. The inlet fitting indexes with the inlet hole at assembly in the structure tube. The inlet fitting has given no trouble and the housing "O" ring has not needed replacing.

The housing has 4 tapped holes for mounting in the structure tube. Slots are provided in the sides of the cylinder for access to the rod/piston pivot pin.

V. PERFORMANCE EVALUATION

The actuator was designed with a 0.006 inch ball stroke. At actuator assembly, the ball stroke was measured as 0.0069 inches. The actuator was mounted in an aluminum tube with 0.2 inch wall thickness. Torsion bars were clamped to the vanes to provide vane loads from 6.4 in-lb/deg to 8.95 in-lb/deg. The nominal load was 7.09 in-lb/deg. The load bars did not apply bending loads to the output shafts.

After the first assembly, the actuator would not operate because the ball had gotten out of the orifice seats and had been clamped between the orifice plates. The ball had been pushed aside by the plunger before the solenoid had clamped the valve assembly together. After backing off the solenoid spring adjustment screw, the ball valve was reassembled, and the normal operation was started by readjustment of the solenoid return spring.

The position feedback pot scale factor was 0.87 V/deg. The measured backlash was 88 mV, or 0.116 deg, which corresponds to a designed backlash of 0.115 deg resulting from approximately 0.0005 inches of clearance in each of the two connecting rod pivot joints. The actuator was controlled with the laboratory actuator control circuit shown in Figure 13. This PDM circuit can provide modulation at 7 carrier frequencies from 50 to 1000 Hz, as well as various types of compensation. For the performance curves in this report the carrier frequency was 150 or 200 Hz, usually 200 Hz. No compensation was used and the gain was about 1.31 V/V as shown in Figure 13. All testing was accomplished with dry air, not nitrogen.

During the first seconds of operation it was apparent that the ball stroke was too large and the carrier frequency was too low. The unloaded vane rates were about 1200 deg/s extending and 2200 deg/s retracting and the PDM carrier frequency was superimposed on the output waveform. The bandwidth was about 50 Hz as observed on the scope. Typical waveforms at 10 and 40 Hz are shown in Figures 14 and 15.

The actuator was operated for about 1 hour, disassembled, and the ball stroke was measured at 0.010 inches due to work hardening of the inlet seat. The spacer was shortened by 0.006 inches to reset the ball stroke to 0.0038 inches, and the carrier frequency was increased to 200 Hz. The output waveforms were much better and the bandwidth at null was slightly over 50 Hz as shown in Figures 16 through 19. All subsequent tests were made with a 200 Hz carrier frequency. The small ($\pm$ 1 deg) signal bandwidth was slightly over 60 Hz, Figure 20. The large signal bandwidth ($\pm$ 5 deg) was 30 Hz, Figure 21. The no load bandwidth was approximately 75 Hz, Figure 22.

There is a large variation between the control cylinder volume at the retracted and extended piston positions due to the short piston stroke. The bandwidth is lower at the extended position than in the retracted position, Figures 23 and 24. The unloaded vane rates were 724 deg/s extending and 833 deg/s retracting, Figure 25.

Previous experience on the Controls Group FOG-M actuators indicated that optimum actuator performance would be attained if the no load vane rates were equal. It was decided to increase the upstream orifice diameter to 0.030 inches and to increase the extending slew rate by 15%. The removal of a broken drill from the upstream orifice necessitated resurfacing of the seat and the use of shims to adjust the ball stroke. Therefore, the ball stroke could not be changed by less than the thickness of 1 shim, or 0.001 inches. Note that one shim therefore represents approximately a $\pm$ 25% change in flow area for a nominal ball stroke of 0.004 inches. The ball valve was reassembled with a measured stroke of 0.0038 inches and a final set of performance tests were made, (see Figures 26 through 31). A final no load slew rate is shown in Figure 32, where the rates are matched within 10%.

It is much more difficult to match the loaded slewing rates due to the difference between choked flow through the vent orifice during retraction and non-chocked flow through the inlet orifice during extension. For small signals, the initial vane rates can be closely matched. When considering aiding hinge moments, it will be difficult to match the vane rates. Vane rates for aiding and opposing torsion loads are shown in Figures 33 and 34. The vane rates for step inputs of 13 degrees are:

Retracting	Opposed	476 deg/s
	Aiding	384
	Average	430 deg/s
Extending	Opposed	283 deg/s
	Aiding	645
	Average	464 deg/s

These curves were measured with the original orifices (0.026 and 0.059) and a ball stroke of 0.004 inches.

The actuator friction was not measured directly but was estimated from the supply pressure required to retract the piston against the stops while loaded with torque bars with a calculated combined spring constant of K = 7.09 in-lb/deg. The piston side forces are highest when the piston is retracting. Assuming that all the friction appears as a piston force opposing piston motion, the friction force is:

$$\mathcal{F} = (P_c A_c - P_s A_r) - \frac{T \cos \theta}{\ell \cos \delta}$$

where

P_c = 14.7 psia

P_s = 1114.7 psia

A_c = .373 in^2

A_r = .198 in^2

θ = 5.3 deg

$$\delta = 15.37 \text{ deg}$$

$$T = K\delta = (7.09)(15.37) = 109 \text{ in-lb}$$

or $$\mathfrak{F} = 9.89 \text{ lb}$$

This piston force corresponds to a torque on the output shaft of:

$$T = \frac{\mathfrak{F}\ell \cos \delta}{\cos \theta} = \frac{(9.89)(.5) \cos (15.37)}{\cos (5.3)}$$

$$T = 4.78 \text{ in-lb}$$

This fricition is 4.4% of the output torque.

The unpressurized piston/actuator friction was estimated by directly measuring the force required to move the piston, which was 3.75 lbs, which corresponds to an output shaft torque of 1.81 in-lb.

The friction is less than the 8% allowed, and the stall torque is greater. The maximum stall torque is:

$$T = \frac{1295 \text{ psig}}{1100 \text{ psig}} (109) = 128.3 \text{ in-lb}$$

The actuator hysteresis and null offset are shown in Figures 35 and 36 for the loaded and unloaded condition. These expanded scales show input vs. output position for a $\pm$ 4 degree input signal.

VI. GAS CONSUMPTION

An open center actuator uses gas from the power supply only on the charge cycle when the piston is extending. Under average conditions, when the control pressure on the piston is about one half the supply pressure, the pressure ratio across the inlet orifice is nearly large enough for some sonic flow to exist. In this case, the sonic flow equation will give good results if the discharge coefficient can be estimated or measured. Assuming $C_d = 1$, and scaling the initial inlet orifice area to accomodate a 0.004 inch stroke and a 0.030 diameter, the maximum flow rate of air (not nitrogen) will be:

$$\dot{w} = \frac{.532 \; C_d \; A_u \; P_s}{T}$$

where: $$A_u \approx (4.4 \times 10^{-4}) \left(\frac{.004}{.006}\right)\left(\frac{.030}{.026}\right) = 3.38 \times 10^{-4} \text{ in}^2$$

$$\dot{w} = \frac{.532(1)(3.38 \times 10^{-4})(1309.7)}{\sqrt{530}} = .0102 \text{ lb/sec}$$

During PDM operation, the ball will be closing the orifice 50% of the time at null conditions, therefore the average flow will be one half the maximum rate:

$$\bar{\dot{w}} = (0.5)\ \dot{w} = (0.5)(.0102) = 5.11 \times 10^{-3} \text{ lb/sec}$$

The orifices in the prototype actuator are nearly identical to those used in the HELLFIRE actuator which has a measured gas concumption rate, including leakage, of 3.102×10^{-3} lb/sec at 550 psig with a 0.029 inch diameter orifice. Scaling the measured HELLFIRE nitrogen rate will give a good estimate of the prototype actuator rate, assuming the same C_d and leakage:

$$\dot{w}_{est} = \left(\begin{matrix} w_{HELL} \\ FIRE \end{matrix}\right)\left(\begin{matrix} STROKE \\ RATIO \end{matrix}\right)\left(\begin{matrix} ORIFICE \\ RATIO \end{matrix}\right)\left(\begin{matrix} P_s \\ RATIO \end{matrix}\right)\left(\begin{matrix} AIR/N_2 \\ RATIO \end{matrix}\right)$$

$$= \left(3.12 \times 10^{-3}\right)\left(\frac{.004}{.006}\right)\left(\frac{.004}{.029}\right)\left(\frac{1295}{550}\right)\left(\frac{.532}{.528}\right)$$

$$\dot{w}_{est} = 5.08 \times 10^{-3} \text{ lb/sec}$$

This is within 1% of the calculated rate at a 50% duty cycle.

VII. CONCLUSIONS

A high performance actuator for a 1.25 inch inside diameter vehicle was designed and tested. The actuator operation was essentially trouble free but the design should be changed for a flight application. Some specific conclusions are:

1. A custom solenoid should be designed, which could be slightly smaller in diameter than the TOW solenoid used in the prototype.

2. The valve seats should be hardened to better withstand the dynamic forces from the ball.

3. The piston should be drilled out on the large end to equalize the response in the extend and retract directions.

4. The designed clearance of 0.0005 inches at each pivot point is acceptable.

5. The piston side forces due to the offset connecting rod are not excessive.

6. The high scale factor on the position feedback potentiometer is acceptable.

7. The standard precision needle bearings have too much clearance and double precision bearings should be used.

8. The actuator bandwidth is approximately 27 Hz when extended and 40 Hz when retracted.

9. A 200 Hz carrier frequency given best performance and "fidelity".

10. The actuator performance was measured on air and without any compensation.

11. The "O" ring gland design on the gas transfer tube inlet is acceptable.

12. The actuator apparently has less friction than anticipated, and greater stall torque.

VIII. RECOMMENDATIONS

Additional testing should be accomplished on this actuator:

1. The piston should be hollowed out and the performance evaluated on nitrogen gas and with appropriate electrical compensation as required.

2. A larger output shaft should be fitted. The larger shaft should include a precision vane attachment design and be tested with a load fixture that also places bending loads on the shaft.

3. Research should be started on an appropriate gas power supply (small squib valve and pressure regulator) to complement the design of the small diameter actuator.

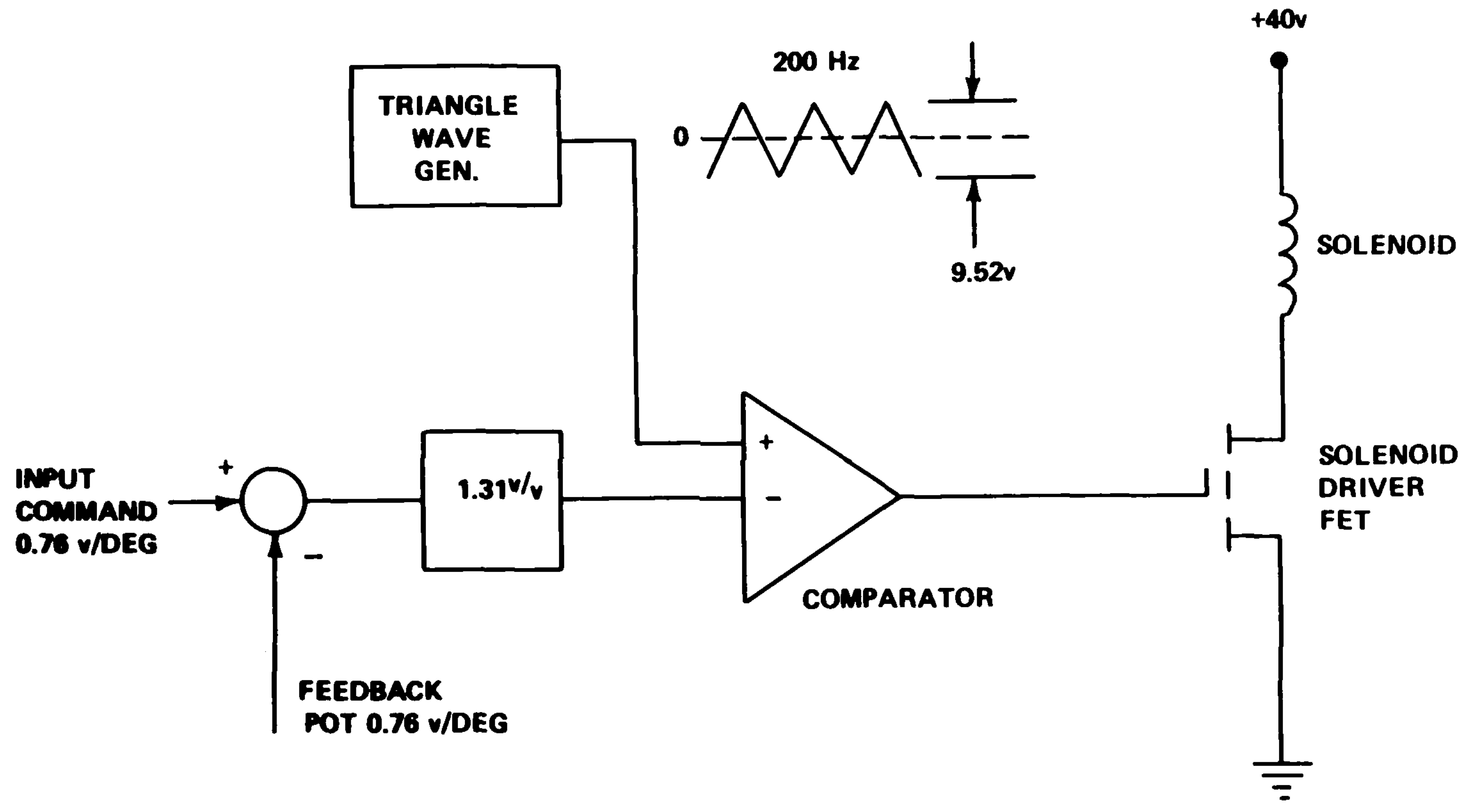


Figure 13. Driving circuit block diagram.

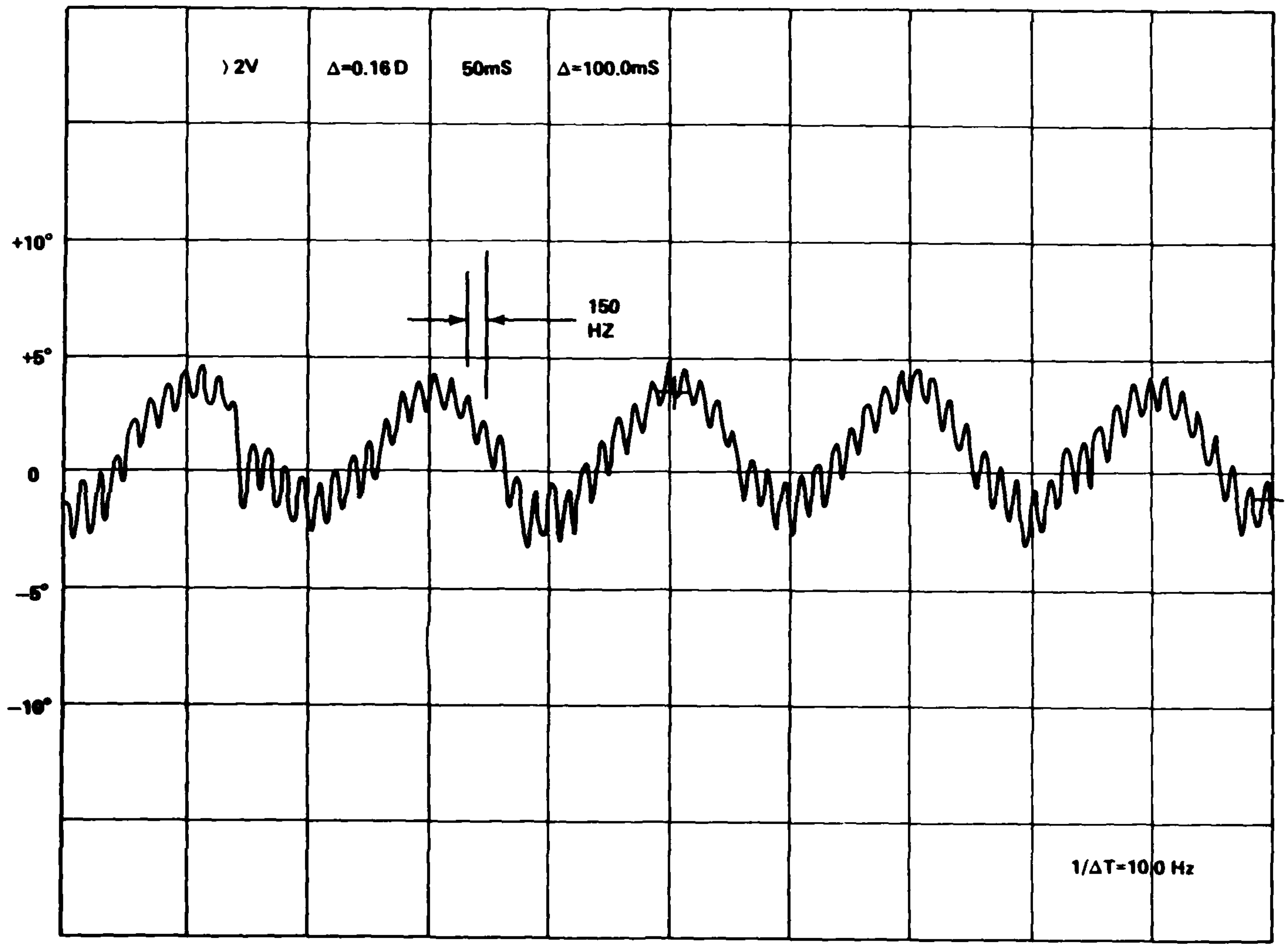


Figure 14. Output waveform, 10 Hz.

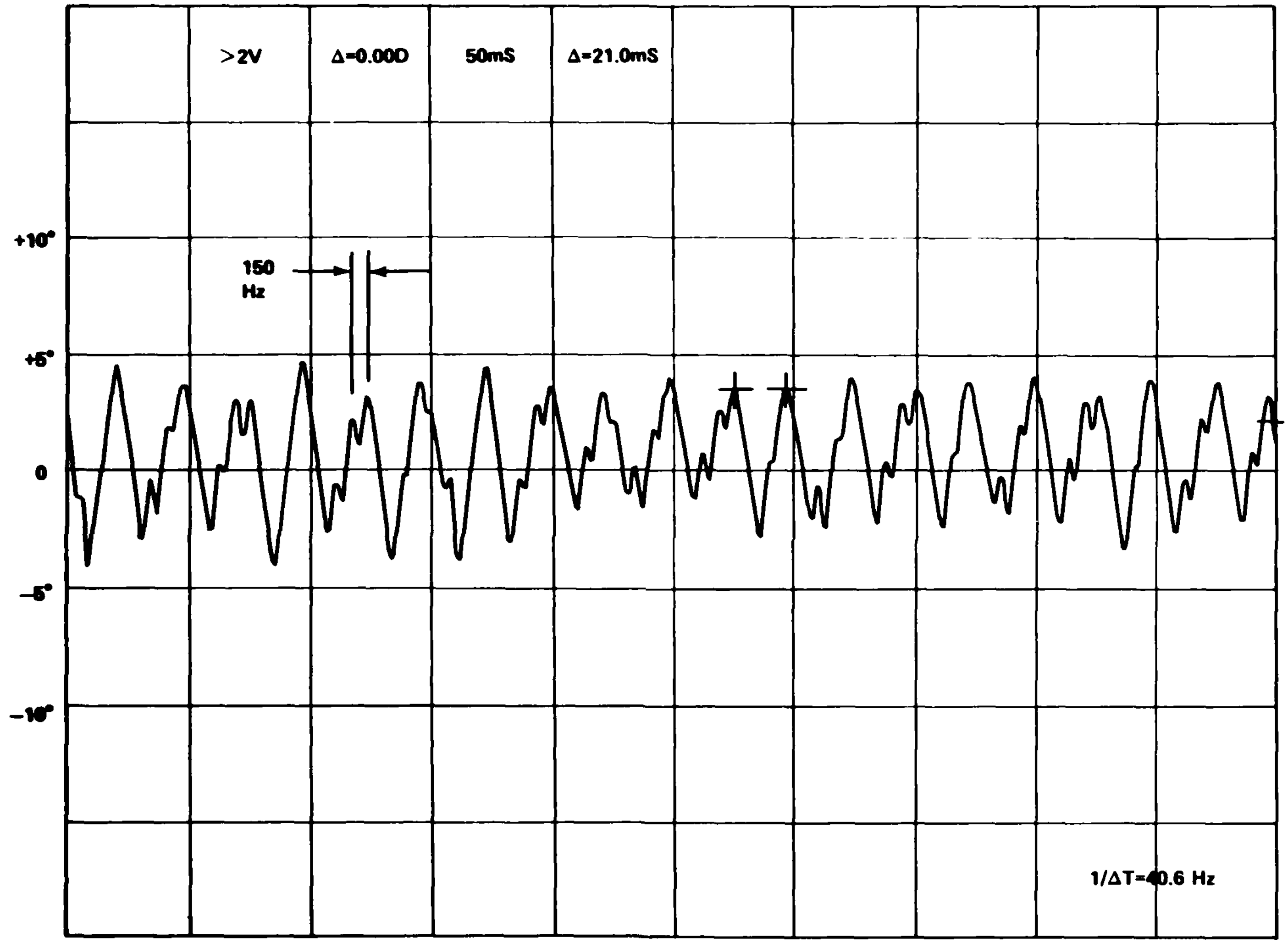


Figure 15. Output waveform, 40 Hz.

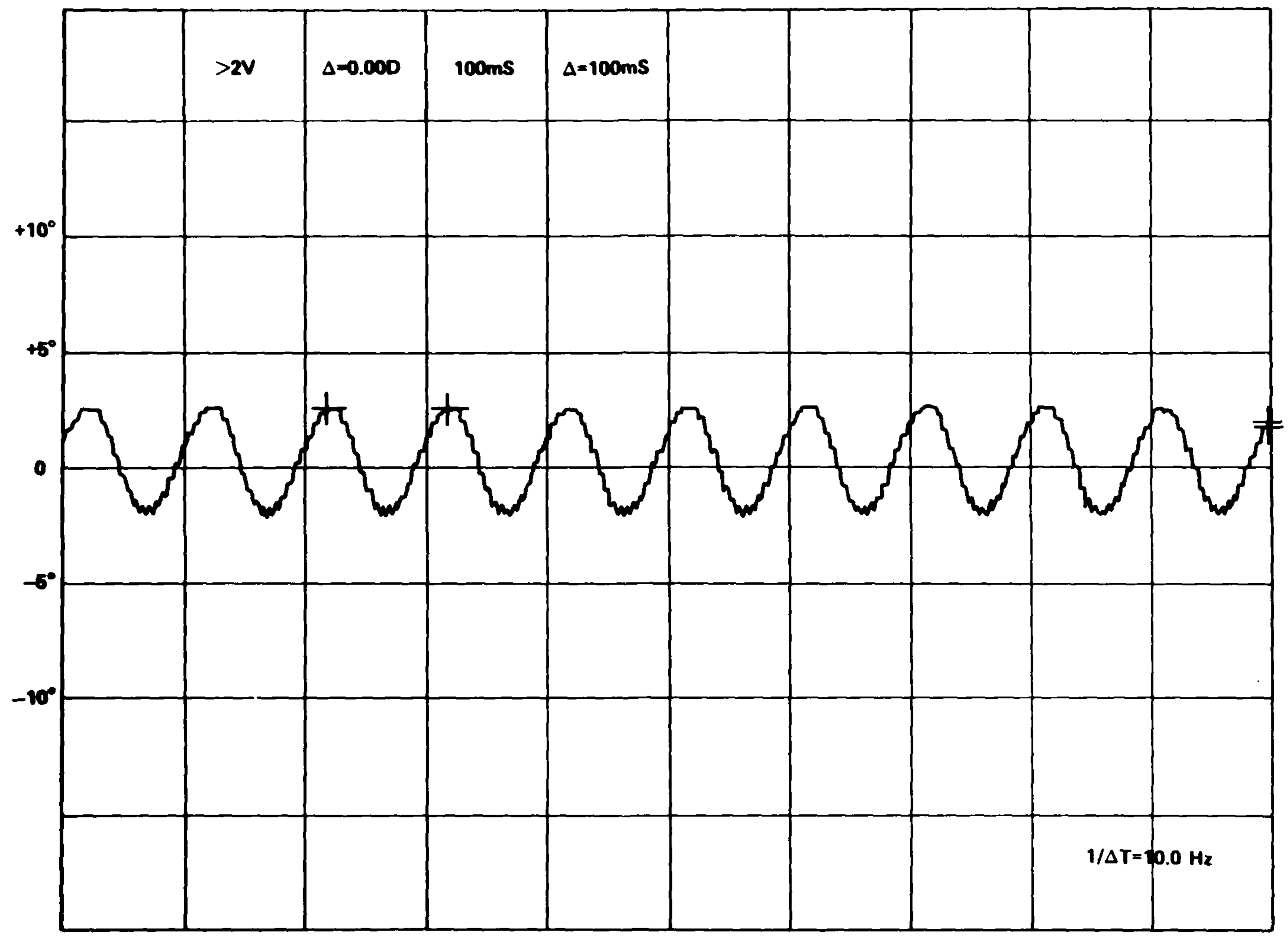


Figure 16. Output waveform, ±2.5 deg at 10 Hz.

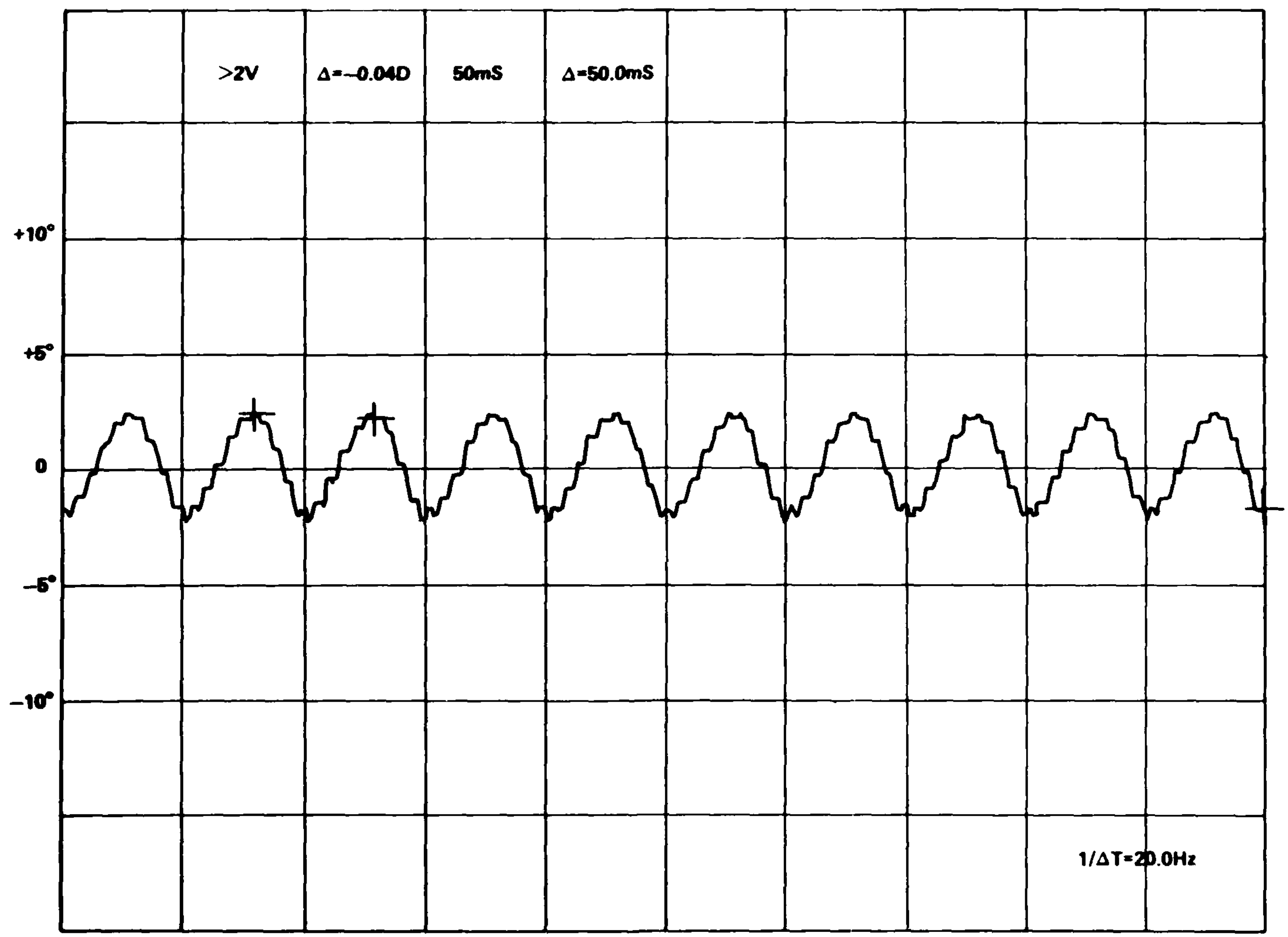


Figure 17. Output waveform, ±2.5 deg at 20 Hz.

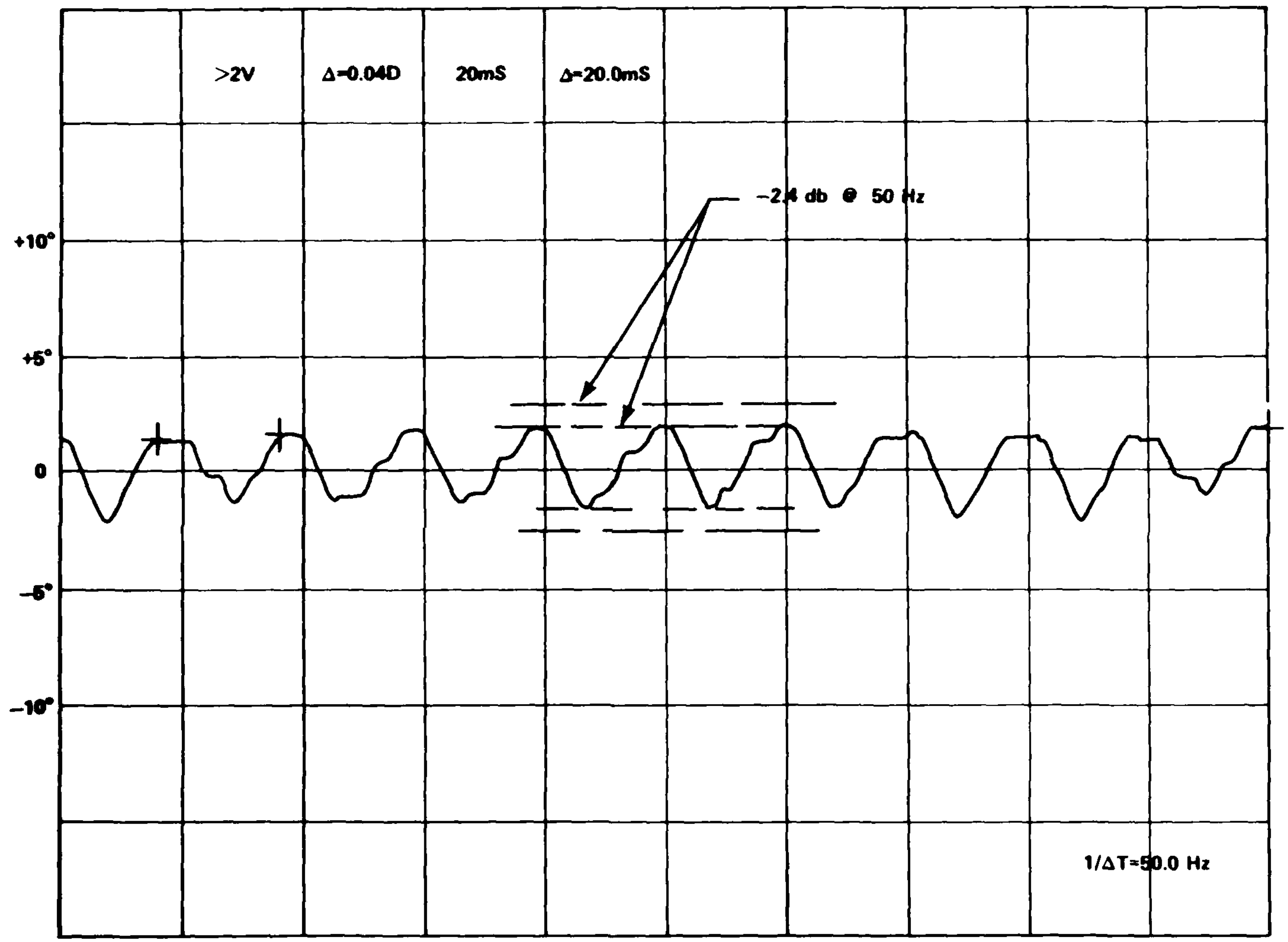


Figure 18. <u>Output waveform, ±2.5 deg at 50 Hz, -2.4 db</u>.

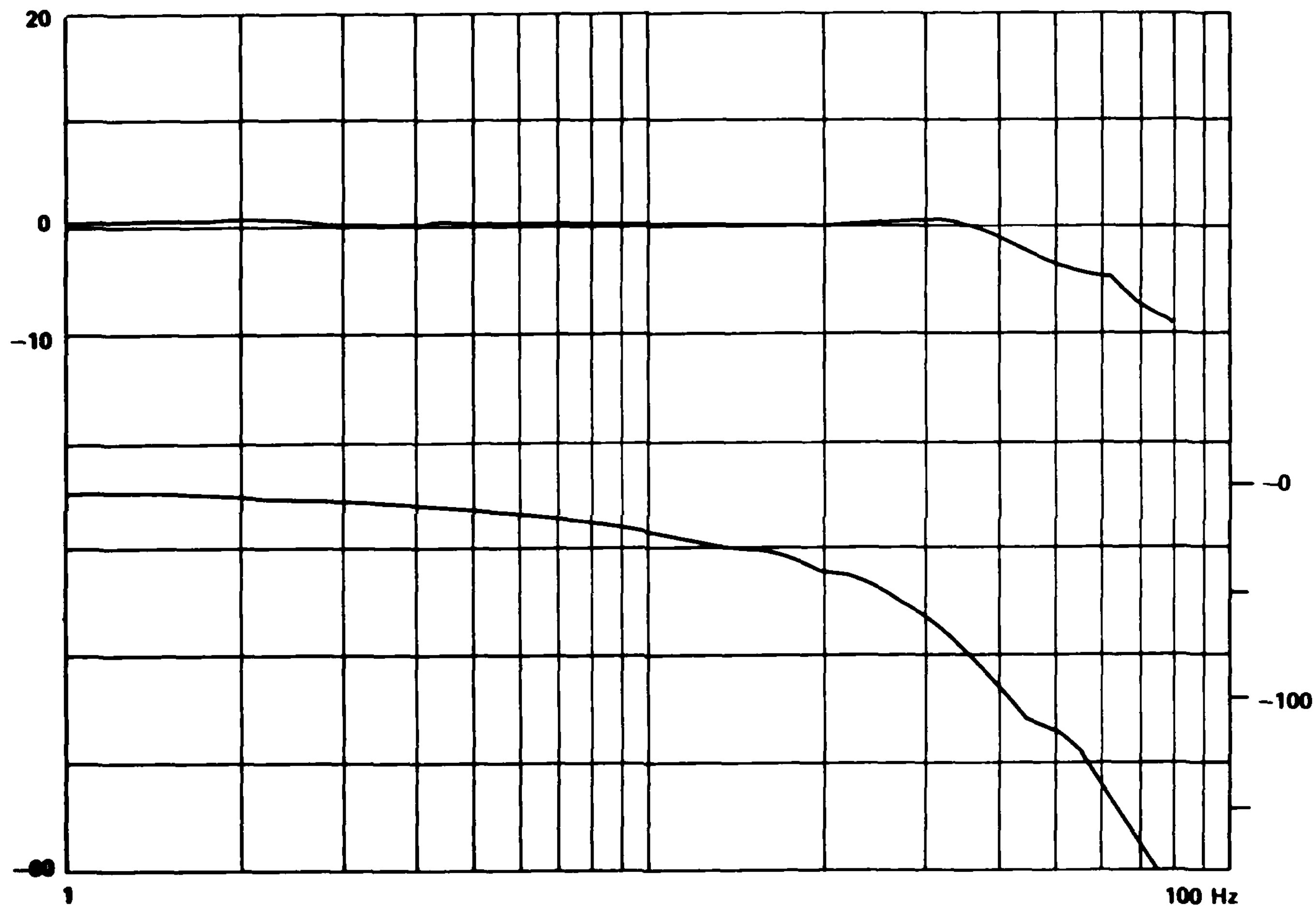


Figure 19. Bode plot, ±2.5 deg at null.

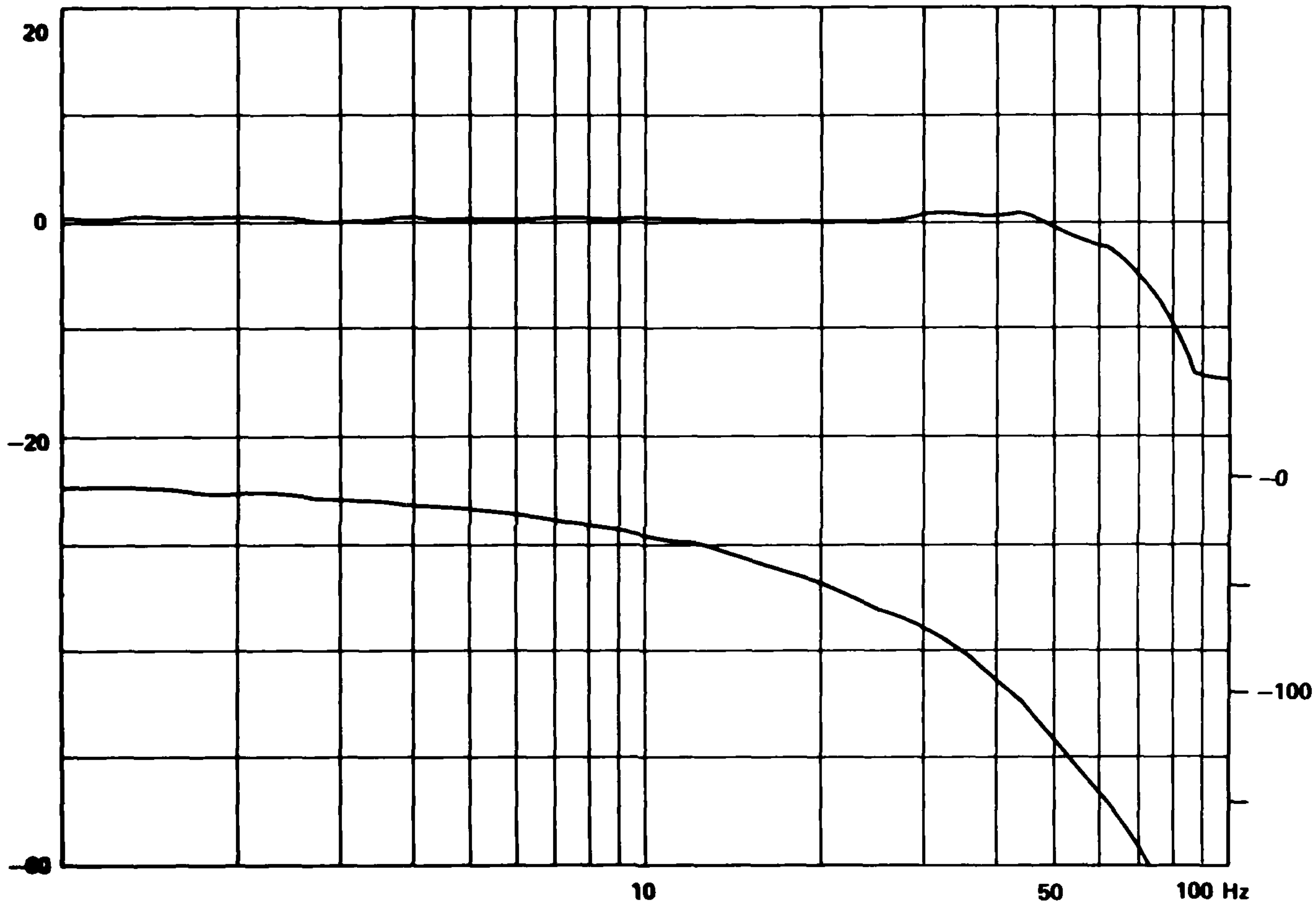


Figure 20. Bode plot, ±1 deg at null.

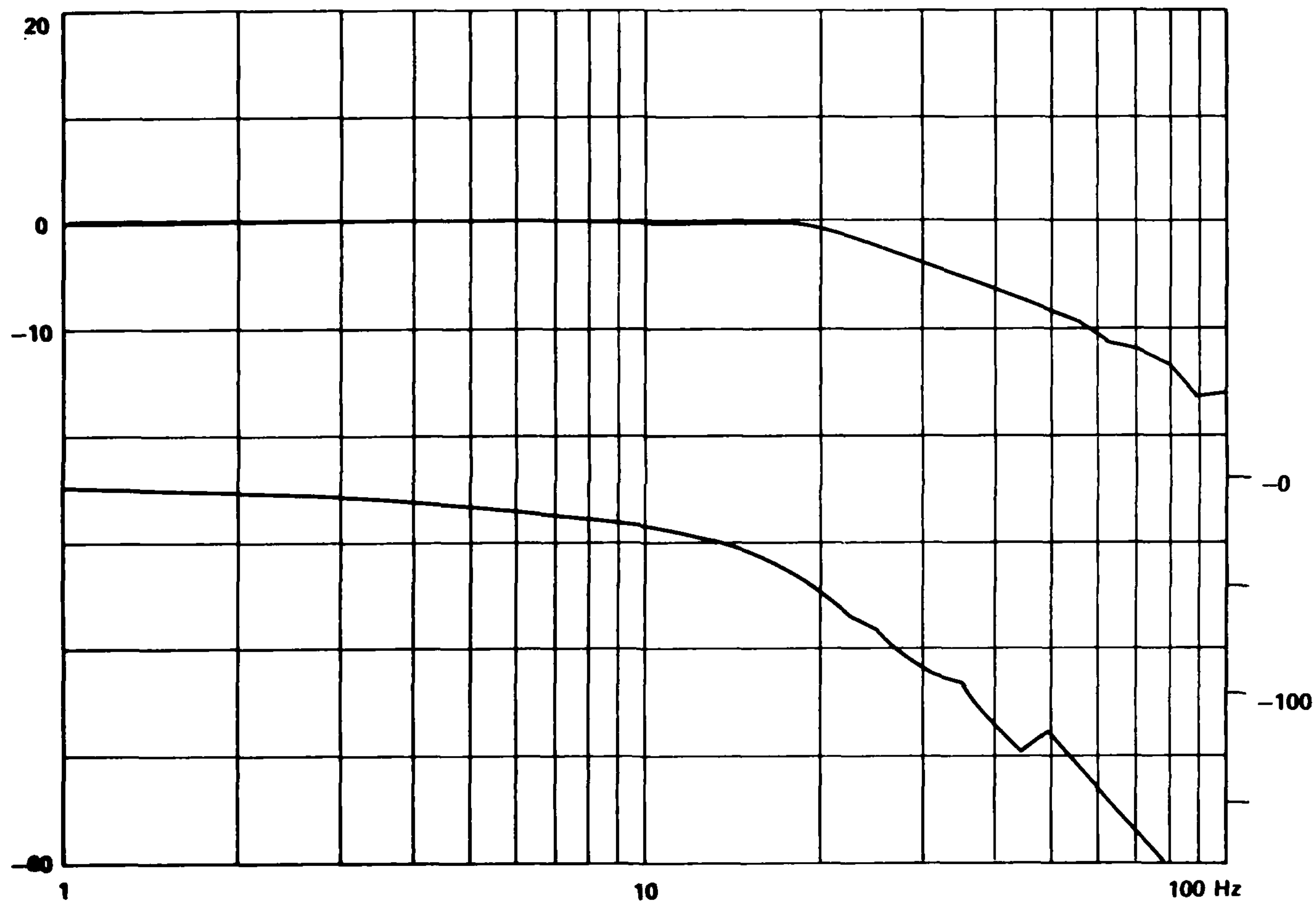


Figure 21. Bode plot, ±5 deg at null.

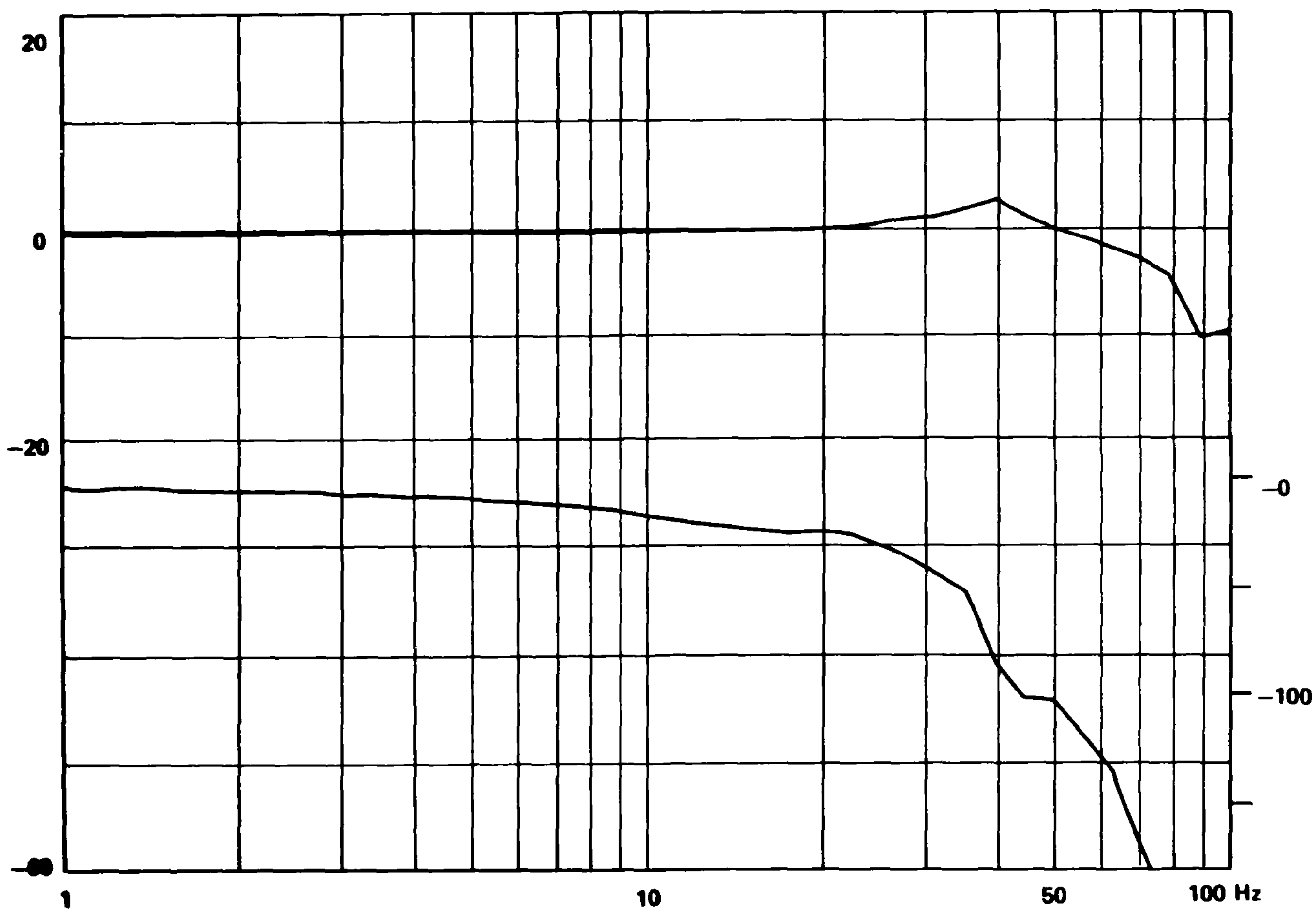


Figure 22. Bode plot, ±2.5 deg at null, no load.

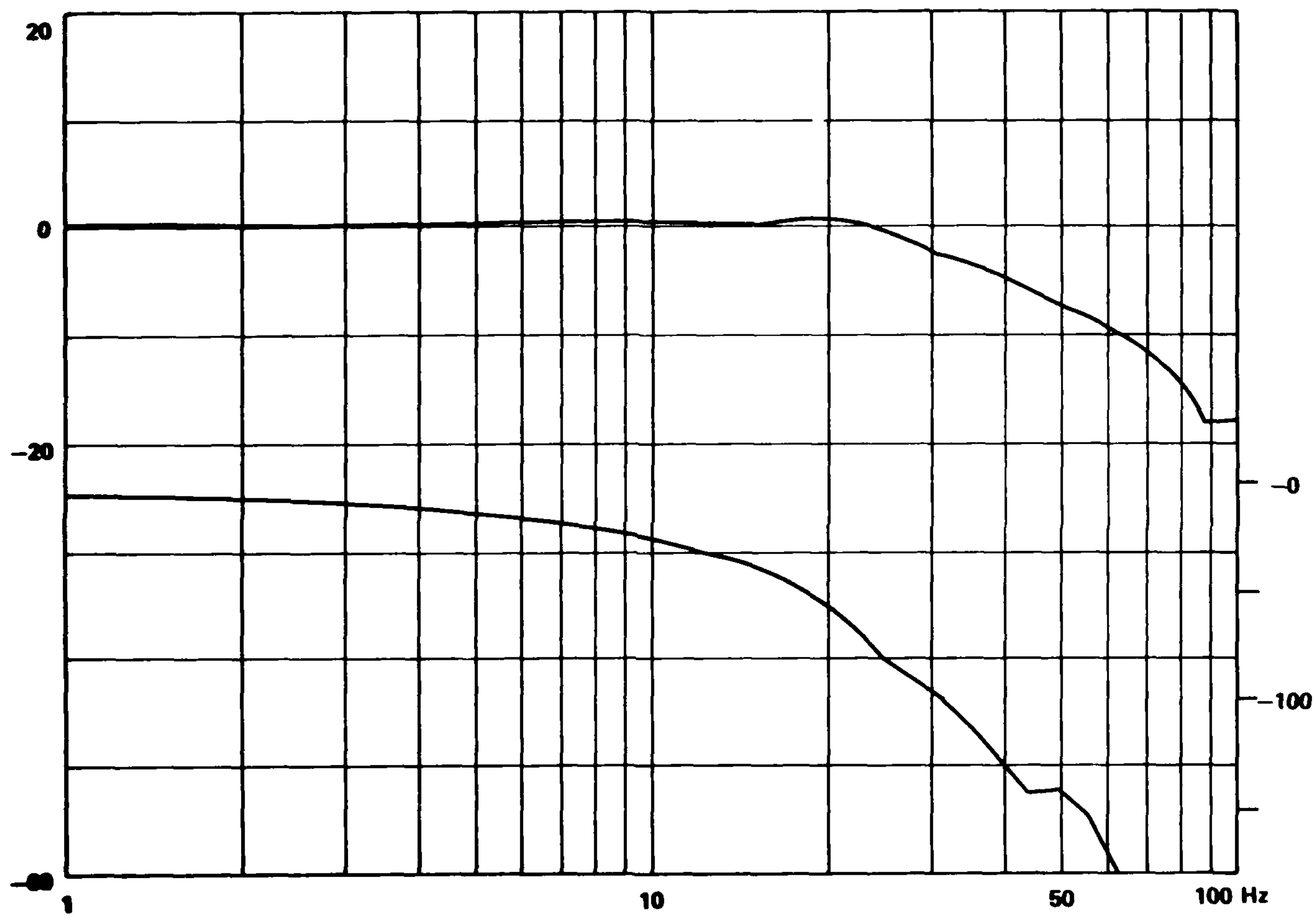


Figure 23. Bode plot, ±2.5 deg at + 7.5 deg, extended.

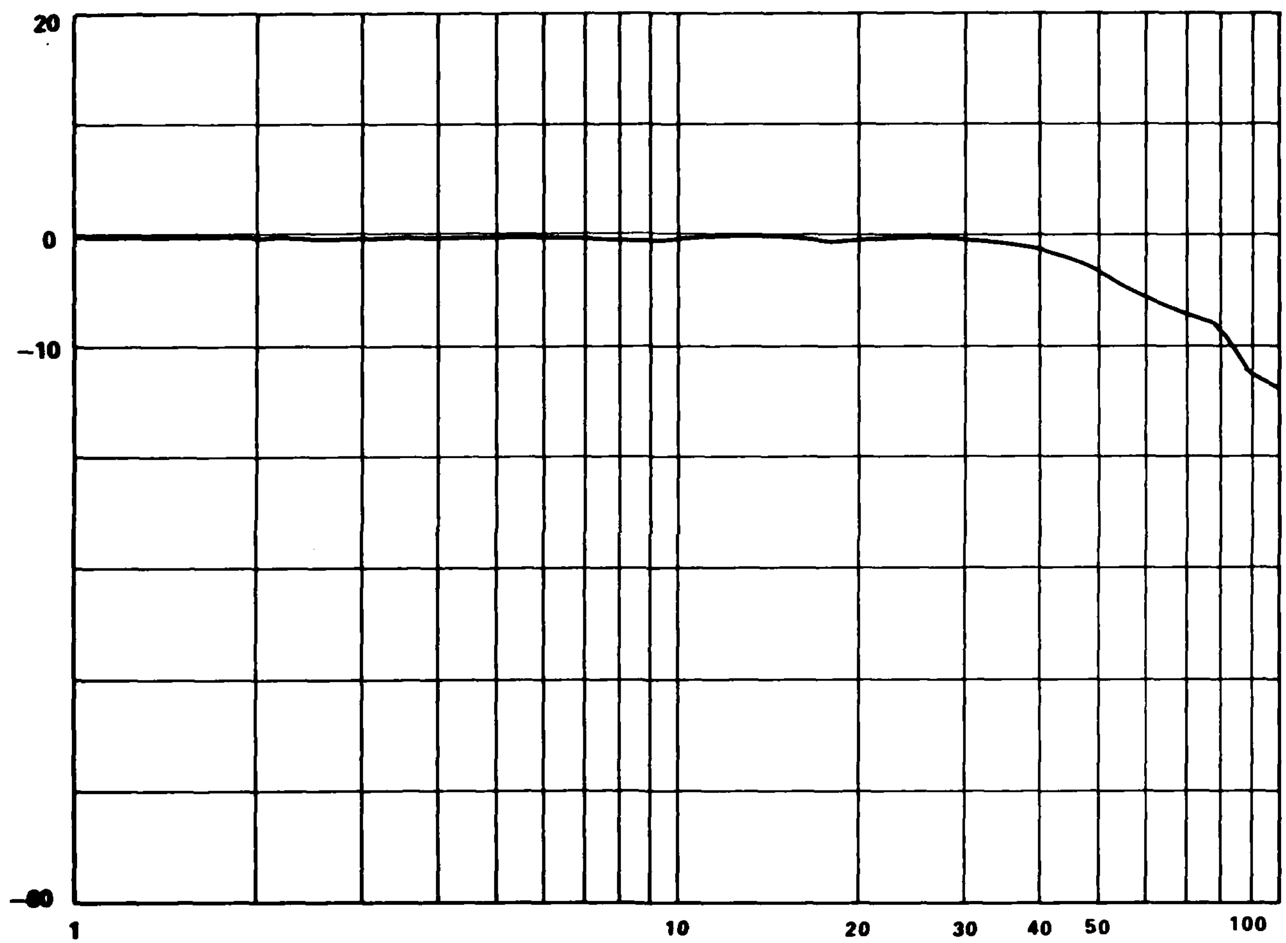


Figure 24. Gain plot, ±2.5 deg at -7.5 deg, retracted.

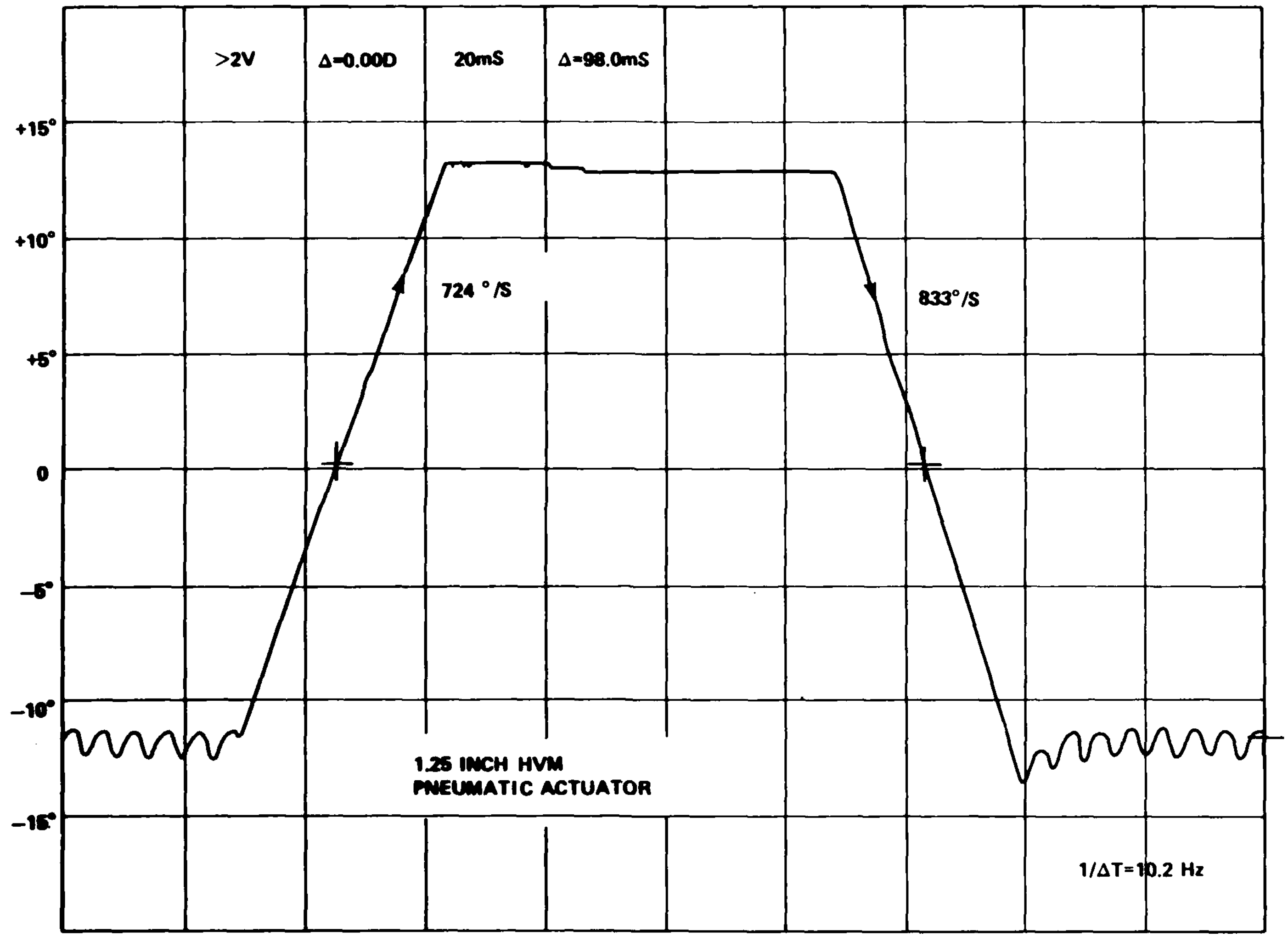


Figure 25. No load slew rates, original orifices.

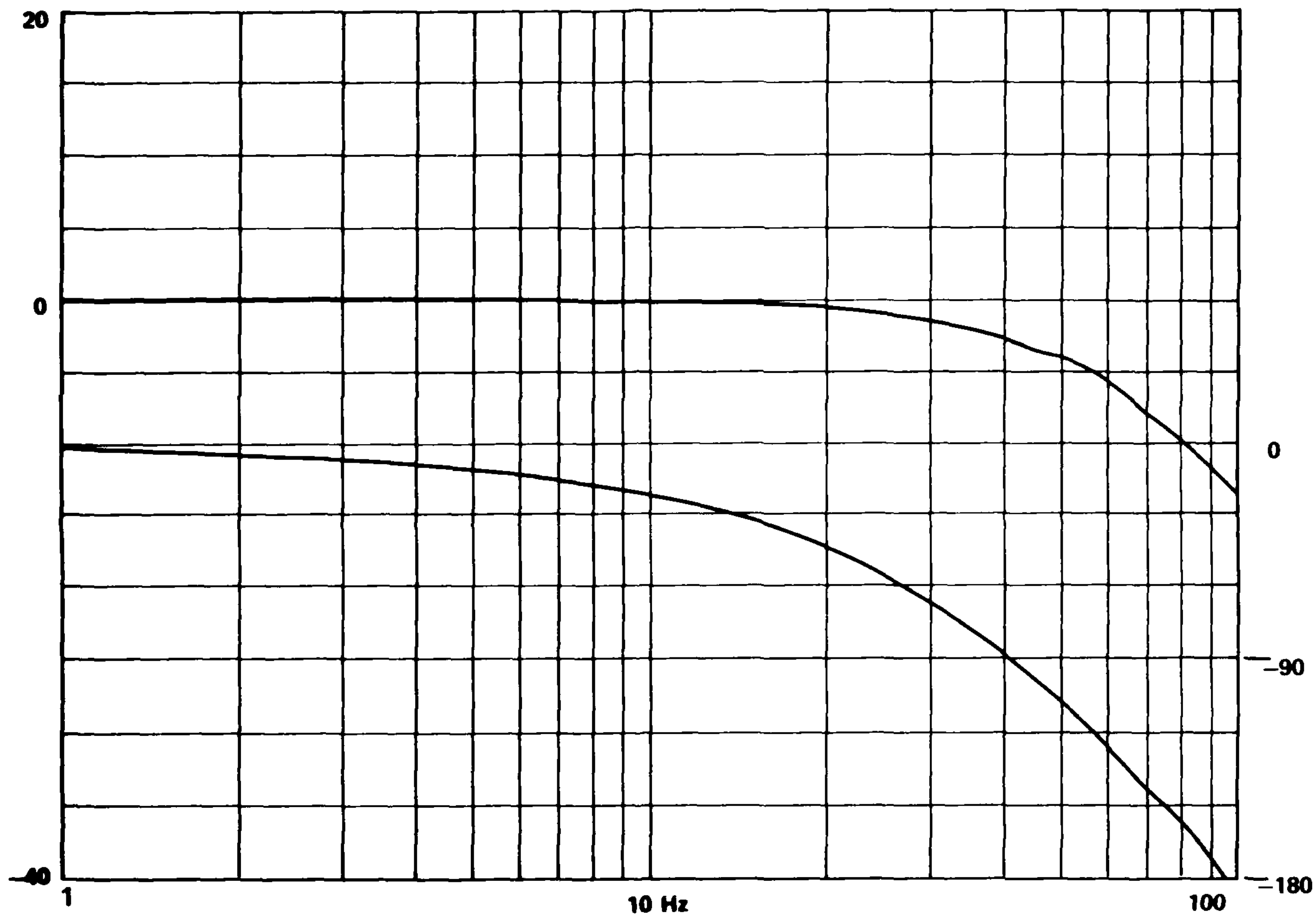


Figure 26. Bode plot, ±1 deg at null.

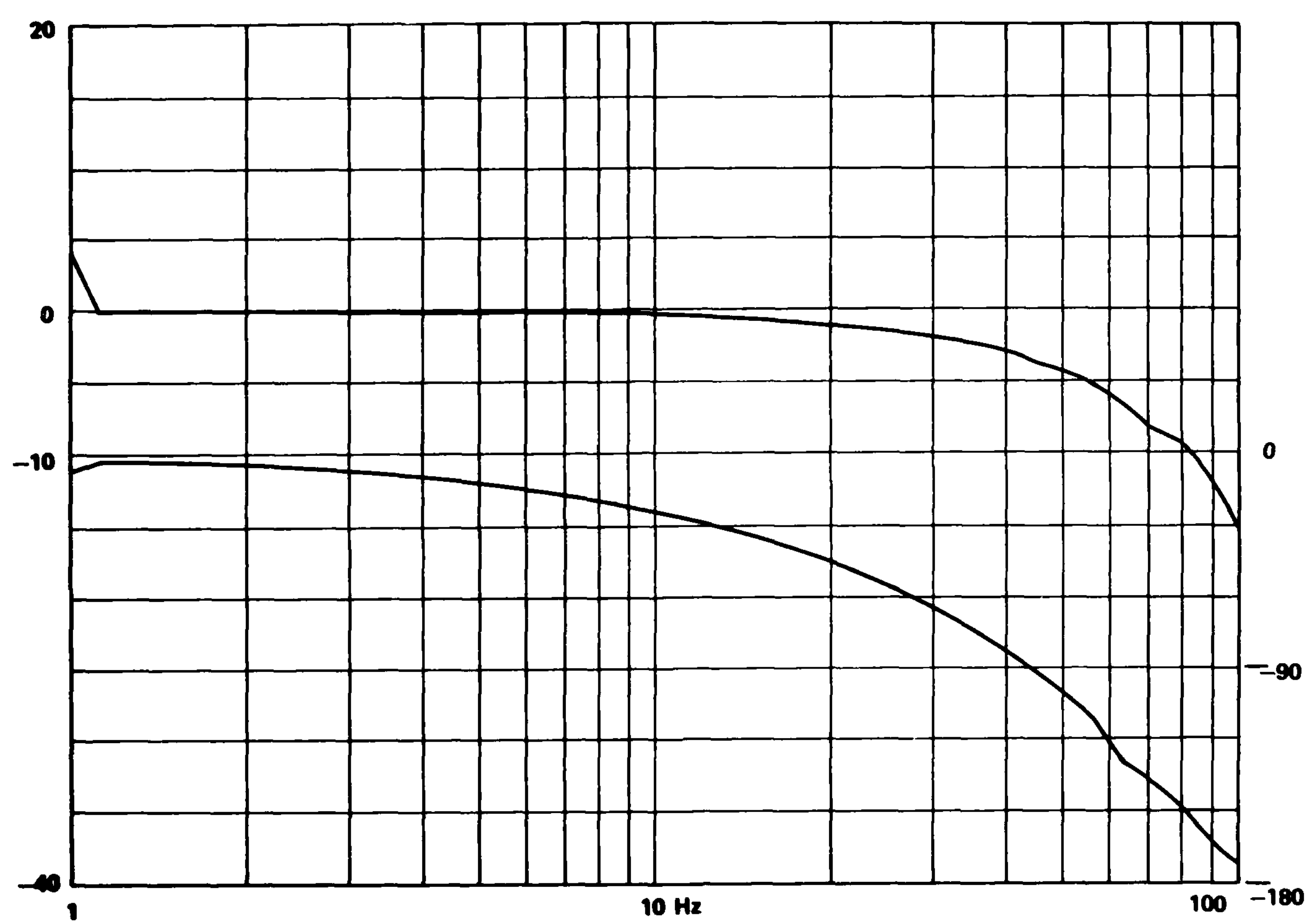


Figure 27. Bode plot, ±2.5 deg at null.

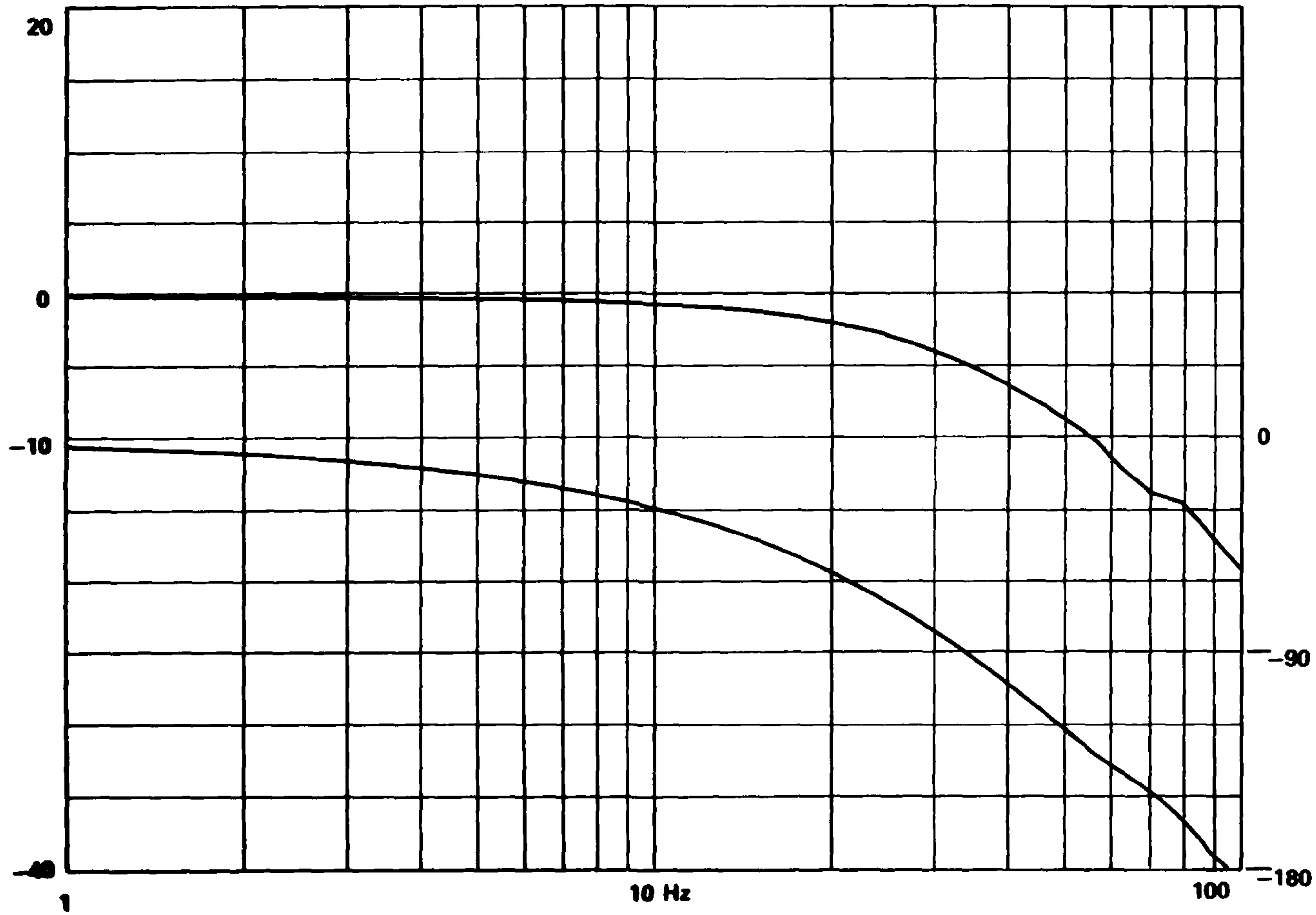


Figure 28. Bode plot, ±2.5 deg at +7.5 deg, extended.

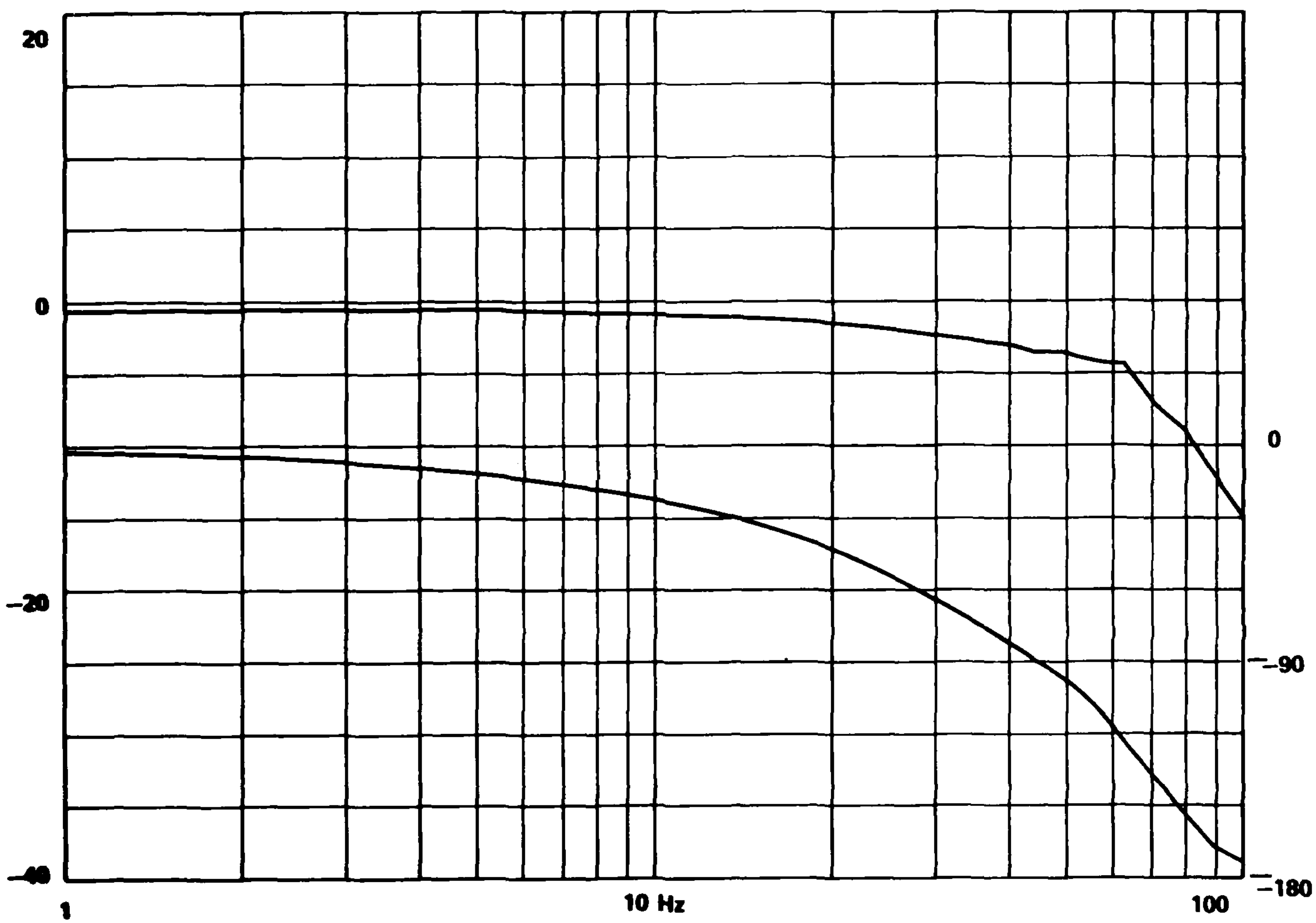


Figure 29. Bode plot, ±2.5 deg at -7.5 deg, retracted.

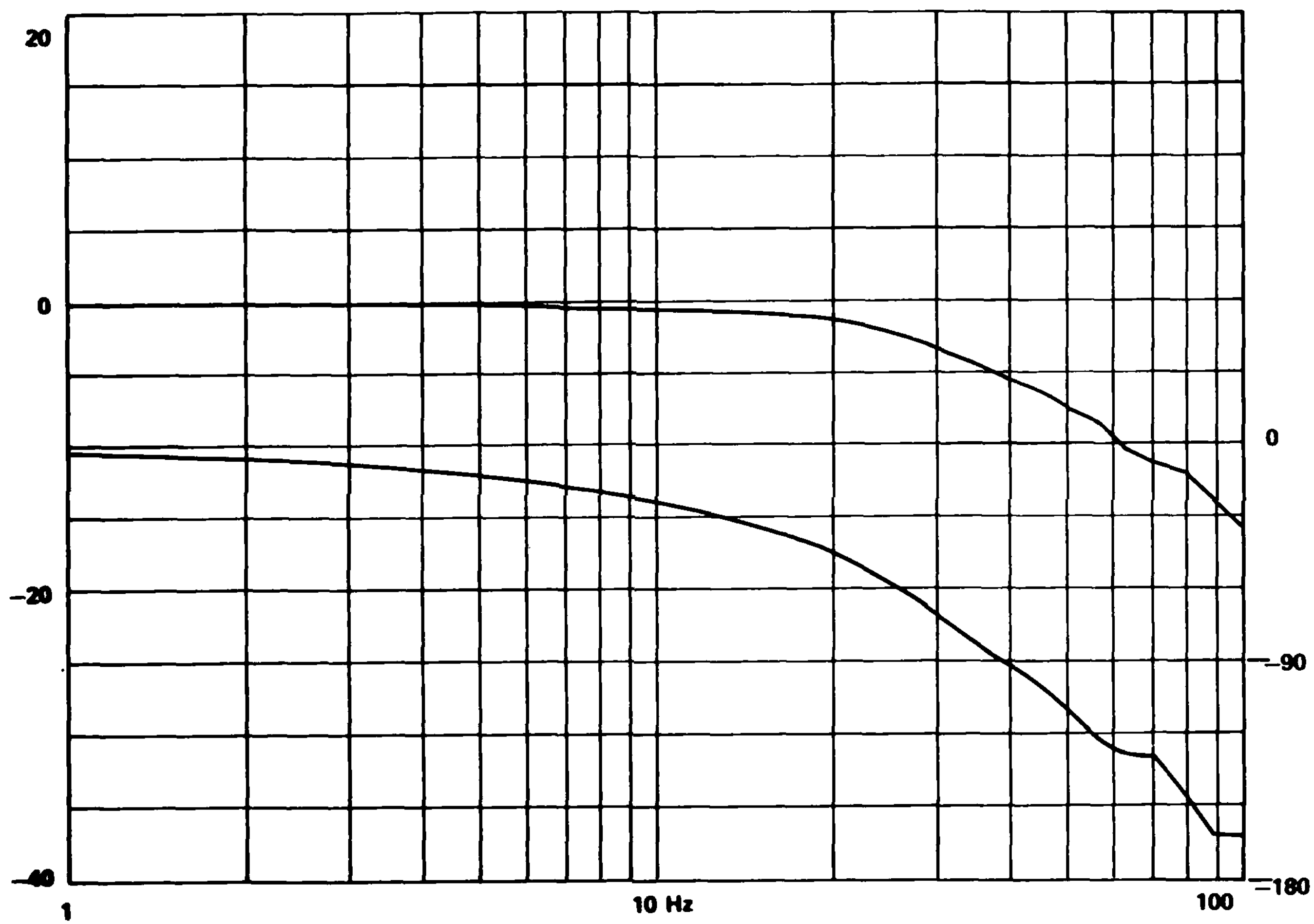


Figure 30. Bode plot, ±5 deg at null.

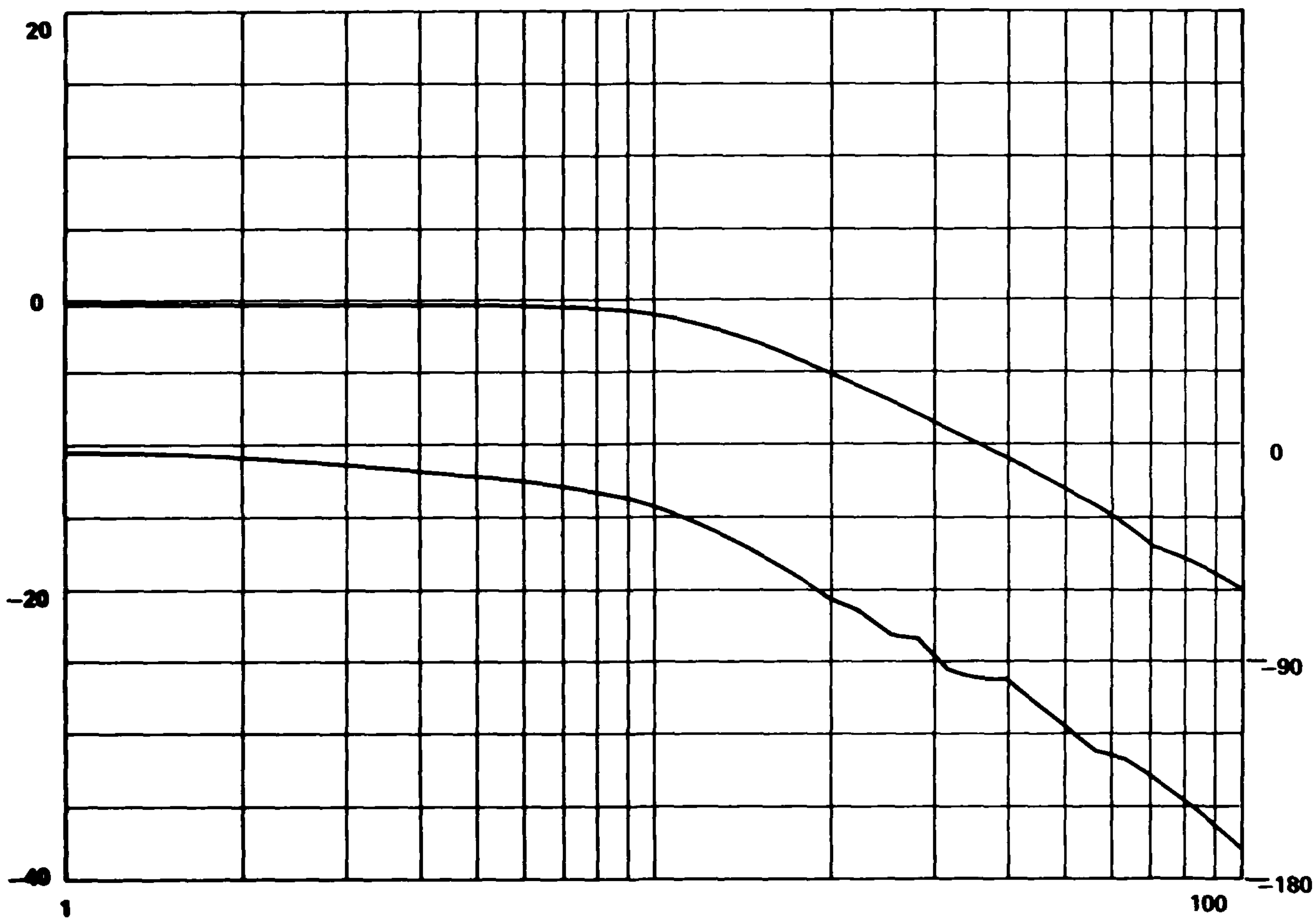


Figure 31. Bode plot, ±10 deg at null.

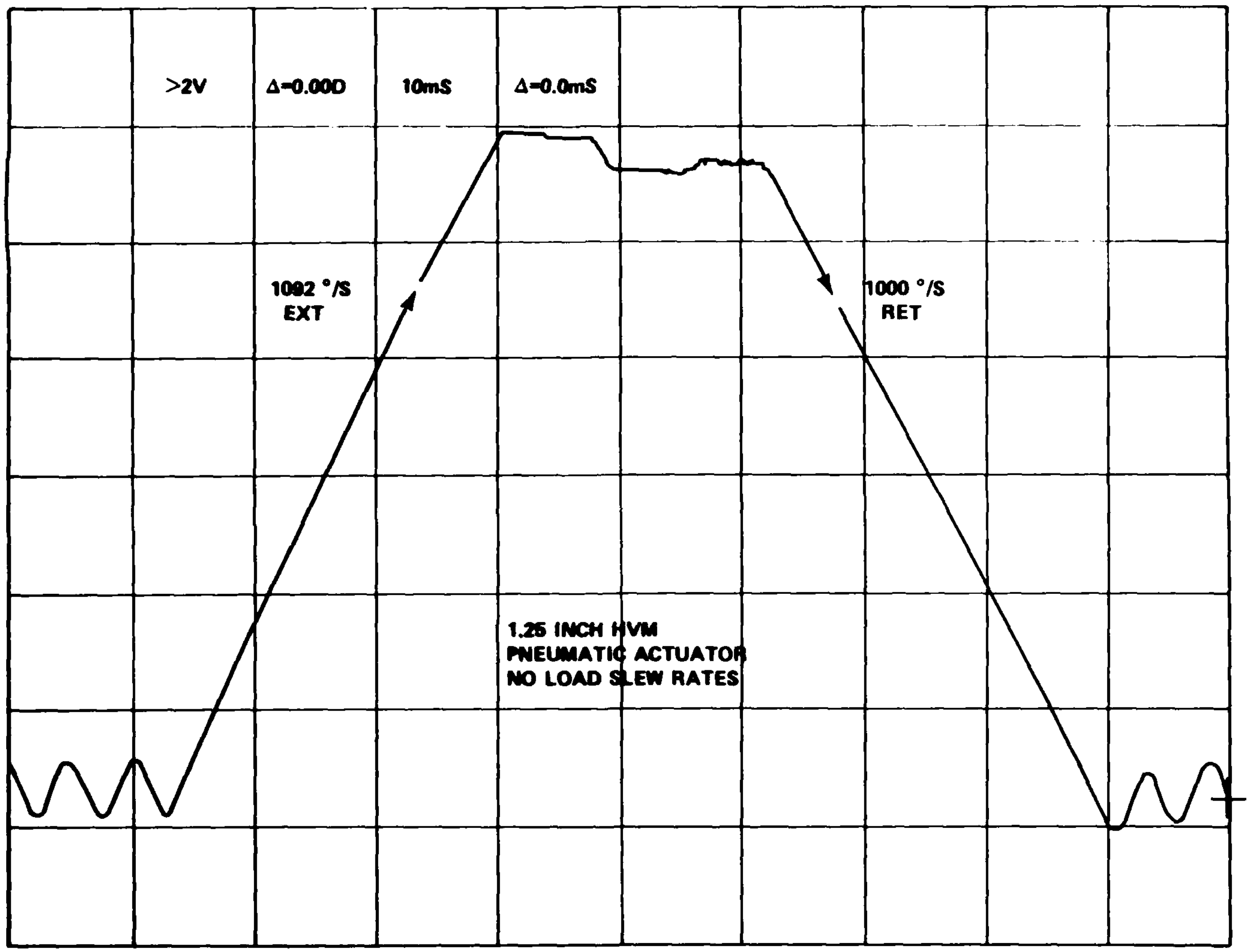


Figure 32. No load slew rates, larger inlet orifice.

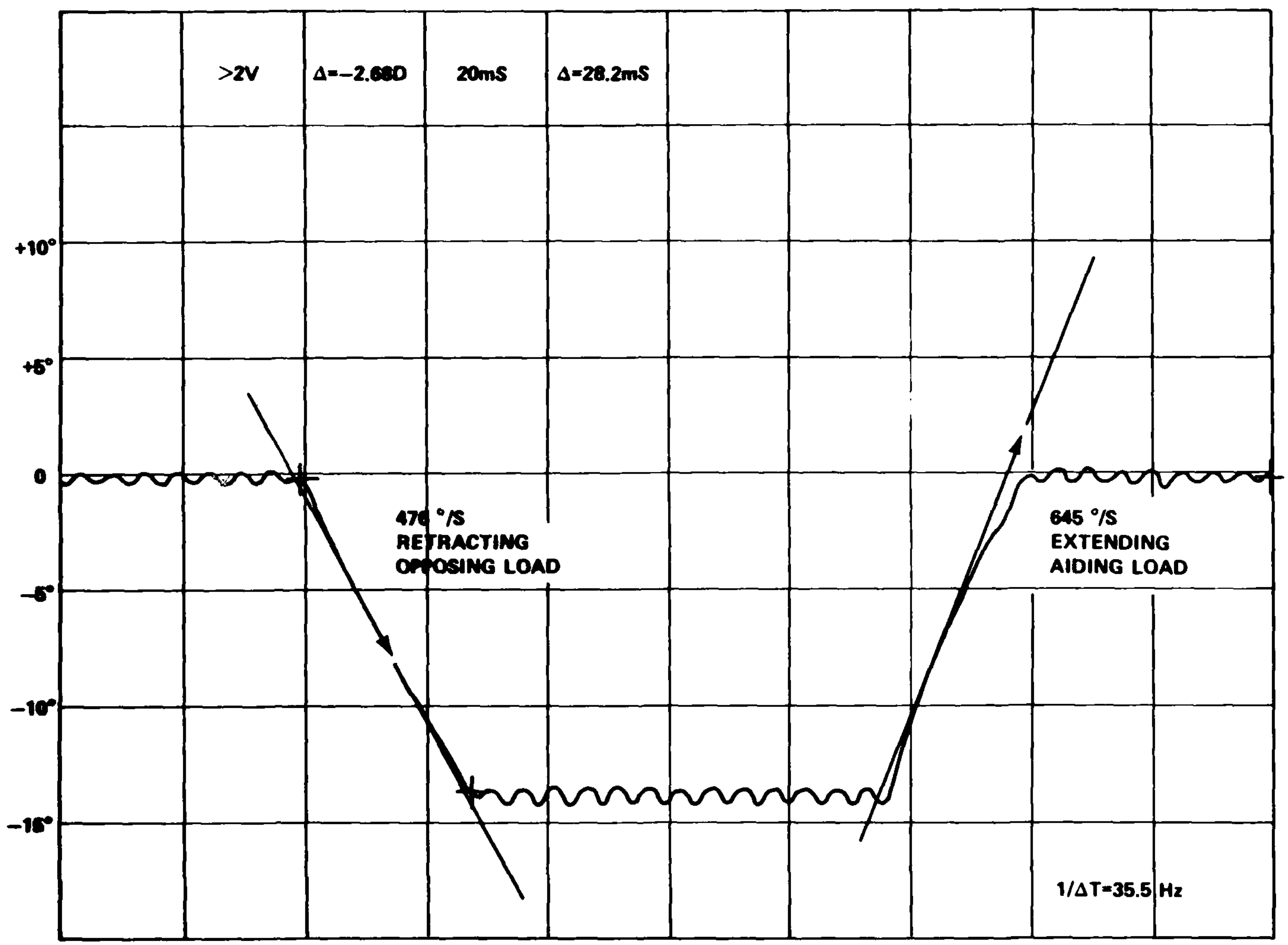


Figure 33. Loaded slew rates.

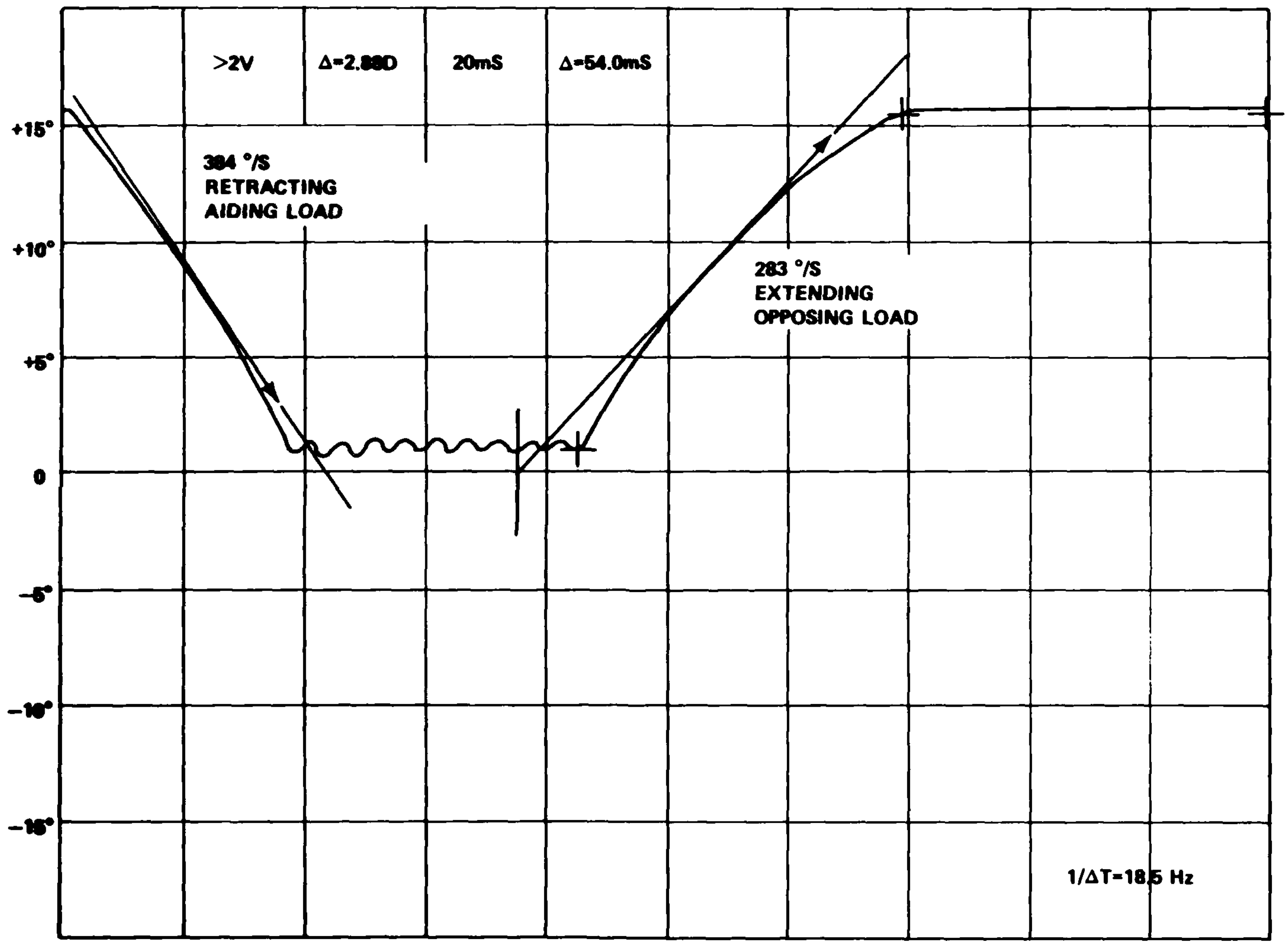


Figure 34. Loaded slew rates.

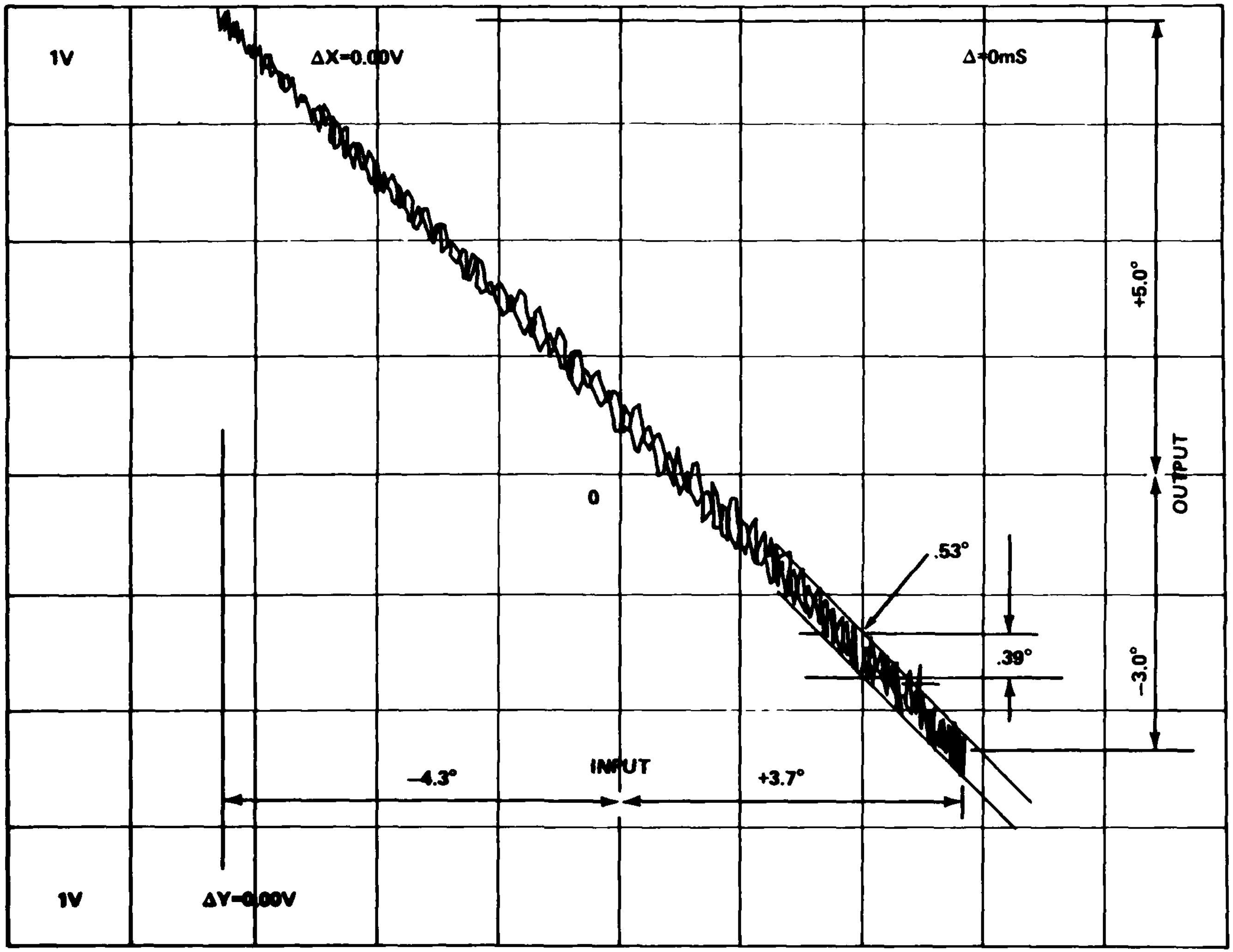


Figure 35. Output position vs commanded position, loaded.

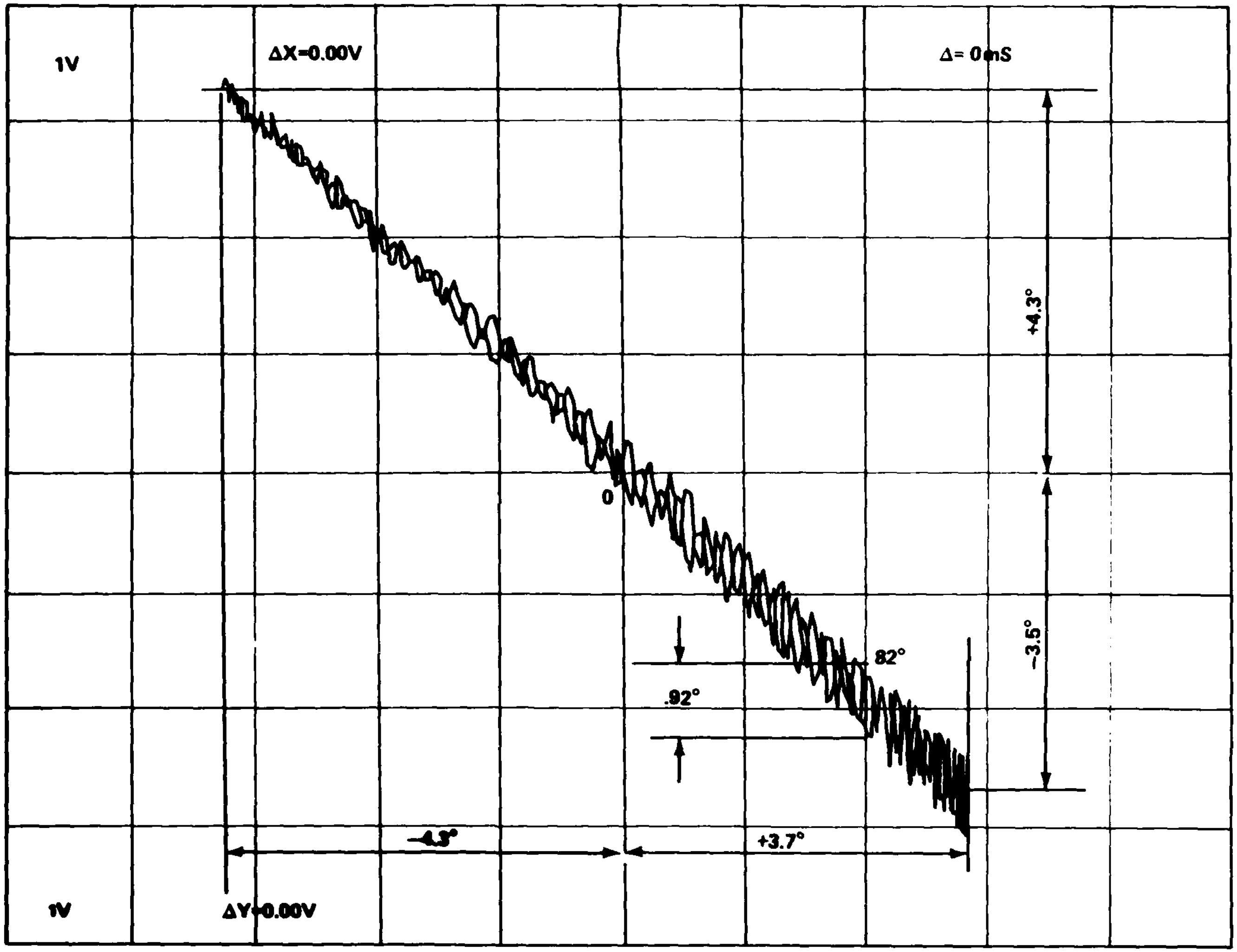


Figure 36. Output position vs commanded position, no load.

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